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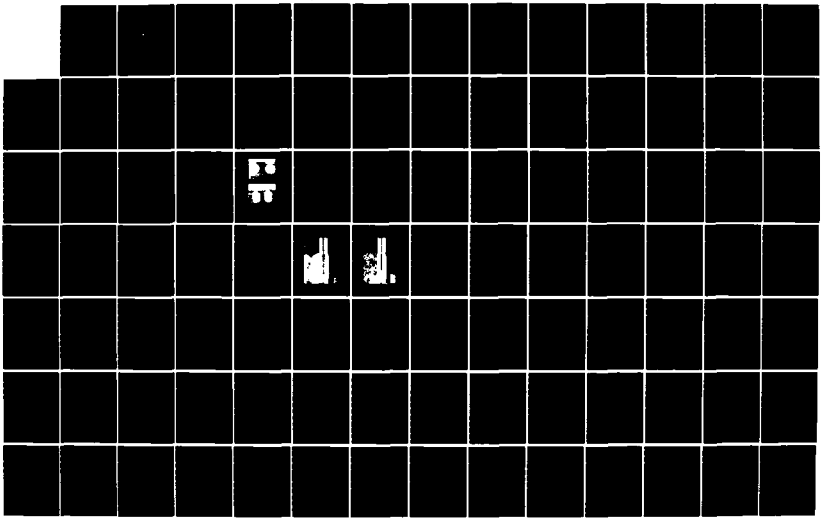

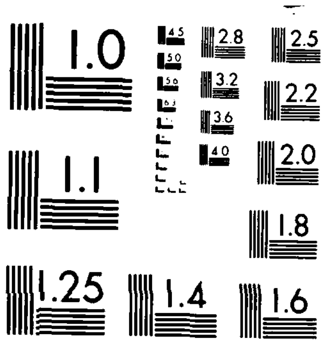

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NAVAL POSTGRADUATE SCHOOL
Monterey, California

THESIS

NUCLEATE POOL BOILING CHARACTERISTICS
OF GEWA-T FINNED SURFACES IN FREON-113

by

Ricardo J. Pulido Q.

September 1984

Thesis Advisor: P. J. Marto

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Pool boiling heat-transfer measurements were made using seven Gewa-T copper tubes in R-113. The first tube (18-mm OD × 0.75 fins/mm × 0.25 mm gap) was tested under three conditions: (a) plain, (b) with four shrouds and (c) with up to 5 wires (0.13 mm diameter) wrapped in the inter-fin trough area. Verifying prior data, the shroud with longitudinal openings of 60° at the top and 8.5° at the bottom gave the best performance among the four

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shrouds tested. This shroud increased the boiling heat-transfer coefficient by 253 percent (over the smooth tube case) at a heat flux of 10,000 W/m^2, while it showed only a 18 percent increase at 100,000 W/m^2. When wire wraps were provided, in all cases, the heat-transfer coefficient was improved at all heat fluxes. The use of three wires gave the best performance with 341-percent increase in heat-transfer coefficient at 10,000 W/m^2 and a 130 percent increase at 100,000 W/m^2.

The next three Fewa-T tubes had dimensions of 25 mm OD × 0.75 fins/mm with fin-tip gaps of 0.15, 0.25 and 0.35 mm, respectively. The last three Gewa-T tubes had the same dimensions except that the fin density was 1.02 fins/mm. For each of these fin densities, the 0.25 mm fin-tip gap produced the best boiling performance at all heat fluxes. Also, for a given fin-tip gap, the boiling performance increased with increase in fin density. For a fin-tip gap of 0.25 mm, the lower fin-density tube produced an 80-percent increase, while the higher fin density tube produced a 103-percent increase in the boiling heat-transfer coefficient.

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Nucleate Pool Boiling Characteristics
of Gewa-T Finned Surfaces in Freon-113

by

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Submitted in partial fulfillment of the
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MASTER OF SCIENCE IN MECHANICAL ENGINEERING

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ABSTRACT

Pool boiling heat-transfer measurements were made using seven Gewa-T copper tubes in R-113. The first tube (19-mm OD × 0.75 fins/mm × 0.25 mm gap) was tested under three conditions: (a) plain, (b) with four shrouds and (c) with up to 5 wires (0.13 mm diameter) wrapped in the inter-fin trough area. Verifying prior data, the shroud with longitudinal openings of 60° at the top and 8.5° at the bottom gave the best performance among the four shrouds tested. This shroud increased the boiling heat-transfer coefficient by 253 percent (over the smooth-tube case) at a heat flux of 10,000 W/m^2 while it showed only a 18 percent increase at 100,000 W/m^2. When wire wraps were provided, in all cases, the heat-transfer coefficient was improved at all heat fluxes. The use of three wires gave the best performance with 341-percent increase in heat-transfer coefficient at 10,000 W/m^2 and a 130 percent increase at 100,000 W/m^2.

The next three Gewa-T tubes had dimensions of 25 mm OD × 0.75 fins/mm with fin-tip gaps of 0.15, 0.25 and 0.35 mm, respectively. The last three Gewa-T tubes had the same dimensions except that the fin density was 1.02 fins/mm. For each of these fin densities, the 0.25 mm fin-tip gap produced the best boiling performance at all heat fluxes. Also, for a given fin-tip gap, the boiling performance

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NOMENCLATURE

$(T_W - T_{SAT})$	average wall superheat (K)
T_B	temperature at the base of the fins (°C)
T_{SAT}	fluid saturation temperature (°C)
T_{AVG}	average wall temperature (°C)
Q_H	heat-transfer rate from the cartridge heater (W)
Q_F	heat-transfer rate through unenhanced end (W)
Q_{LOSS}	total heat-loss rate (W)
q	heat flux from the Gewa-T surface (W/m^2)
$\bar{h}$	average heat-transfer coefficient ($W/m^2.K$)
k_C	thermal conductivity of the copper (W/m.K)
k	thermal conductivity of Freon-113 liquid (W/m.K)
β	volumetric thermal expansion coefficient (1/K)
ν	kinematic viscosity (m^2/s)
α	thermal diffusivity (m^2/s)
ρ	density of Freon-113 (kg/m^3)
Nu	Nussel number
Ra	Rayleigh number
Pr	Prandtl number
V	voltage across the cartridge heater element (volts)
V_{RES}	voltage drop across the precision resistor (volts)
I	current through the heater element (amps)

R	resistance of the precision resistor (ohms)
O.D.	outside diameter of the Gewa-T tube (m)
D_1	diameter at the position of the thermocouples (m)
D_2	diameter at the base of the fins (m)
D_o	outside diameter of the unenhanced ends (m)
D_i	inside diameter of the unenhanced ends (m)
P	outside wall perimeter of the unenhanced ends (m)
H_F	fin height (m)
L	length of enhanced test section (m)
L_U	length of unenhanced ends (m)
L_C	corrected length of unenhanced ends (m)
t	tube thickness of unenhanced ends (m)
A_B	surface area at the base of the fins (m^2)
A_C	cross-sectional area of tube wall at the unenhanced ends (m^2)
A_S	outside surface area of the unenhanced ends (m^2)
g	gravitational acceleration (m/s^2)

excluded), and the two thermocouples for vapor and liquid temperature measurements were calibrated in an insulated, stainless-steel bath. The tubes were completely immersed into the bath, which was provided with a motor-driven mixer. The bath temperature was measured using a platinum-resistance thermometer accurate to ±0.01 K.

Since the temperature measurements during this investigation ranged from 45°C to 60°C, thermocouple calibration was performed in the range of 45°C to 70°C. To begin the calibration process, hot water (at about 75°C) was added to the insulated container. Following proper mixing, all thermocouple readings were automatically recorded through the data-acquisition system. The bath temperature was measured using the platinum-resistance thermometer. To obtain other lower temperatures, small quantities of water were added to the bath, allowing 5 minutes for temperature stabilization of the test section.

The discrepancy (i.e., the platinum-resistance thermometer reading minus the thermocouple reading) was plotted against the thermocouple reading as shown in Figure 4. The second order polynomial curve shown in this figure was generated using the data from all seventeen thermocouples. However, for clarity, only three thermocouples were selected for data points from each tube. It is evident that the thermocouples in both tubes show similar discrepancies, the thermocouple readings have only about ±0.1 K discrepancy around the

Plexiglas lid, which had been fitted with a rubber O-ring. Several holes were drilled in the lid to fix in place the test section extension, vent tube, connections for primary and secondary condensers, and two more thermocouples to sense both the bulk liquid and vapor temperatures. The Pyrex glass vessel with the attached fittings was placed on the plate heater, which served to maintain the liquid at saturation temperature. A secondary condenser was mounted external to the vessel.

C. INSTRUMENTATION

The power supply was connected to the cartridge heater through a voltage regulator and then to a Variac scaled from 0 to 120 VAC. This arrangement allowed fine adjustment for a stable feed voltage during the runs. A precision resistor of 2.031 ohms was connected in series with the cartridge heater to provide a measure of the current flow by reading the voltage drop across this element. A digital voltmeter was used to measure both the voltage across the resistor and the voltage across the cartridge heater. All the thermocouples were connected to a computer data acquisition system, and ambient conditions and voltage readings were supplied manually. The microcomputer processed all this information through a program included in Appendix A.

D. CALIBRATION OF THERMOCOUPLES

A total of fifteen wall thermocouples in tubes 1 and 7 (one thermocouple in tube 1 was defective, hence it was

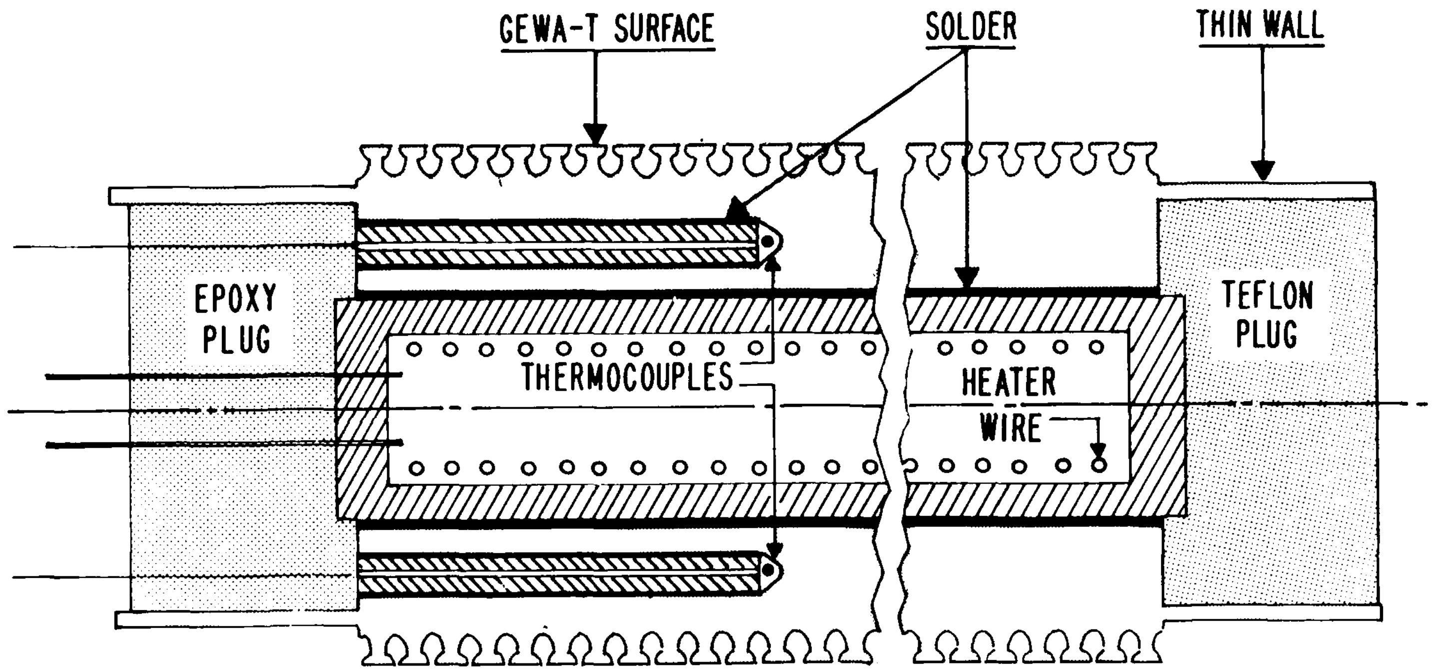

Figure 3. Cross-Sectional View of Test Section

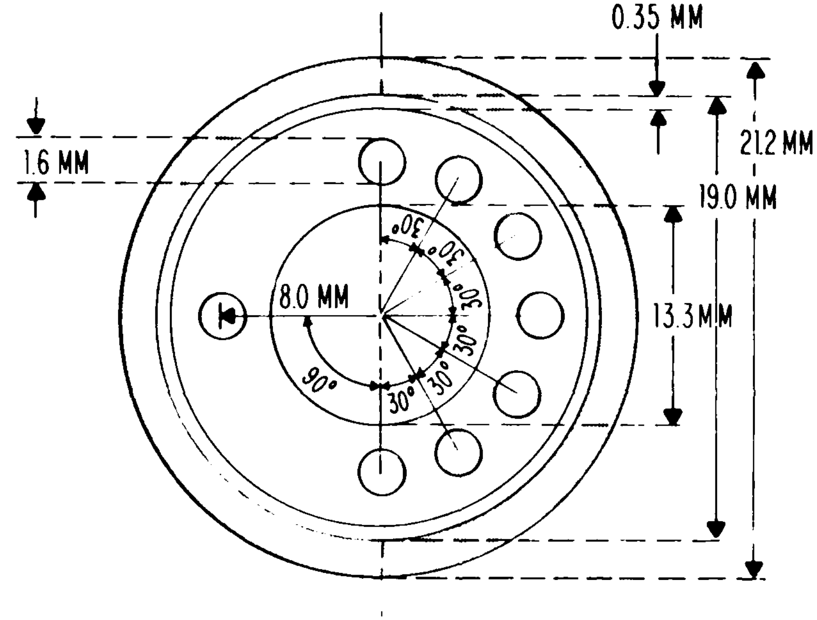

Figure 2. Thermocouple Radial and Angular Positions

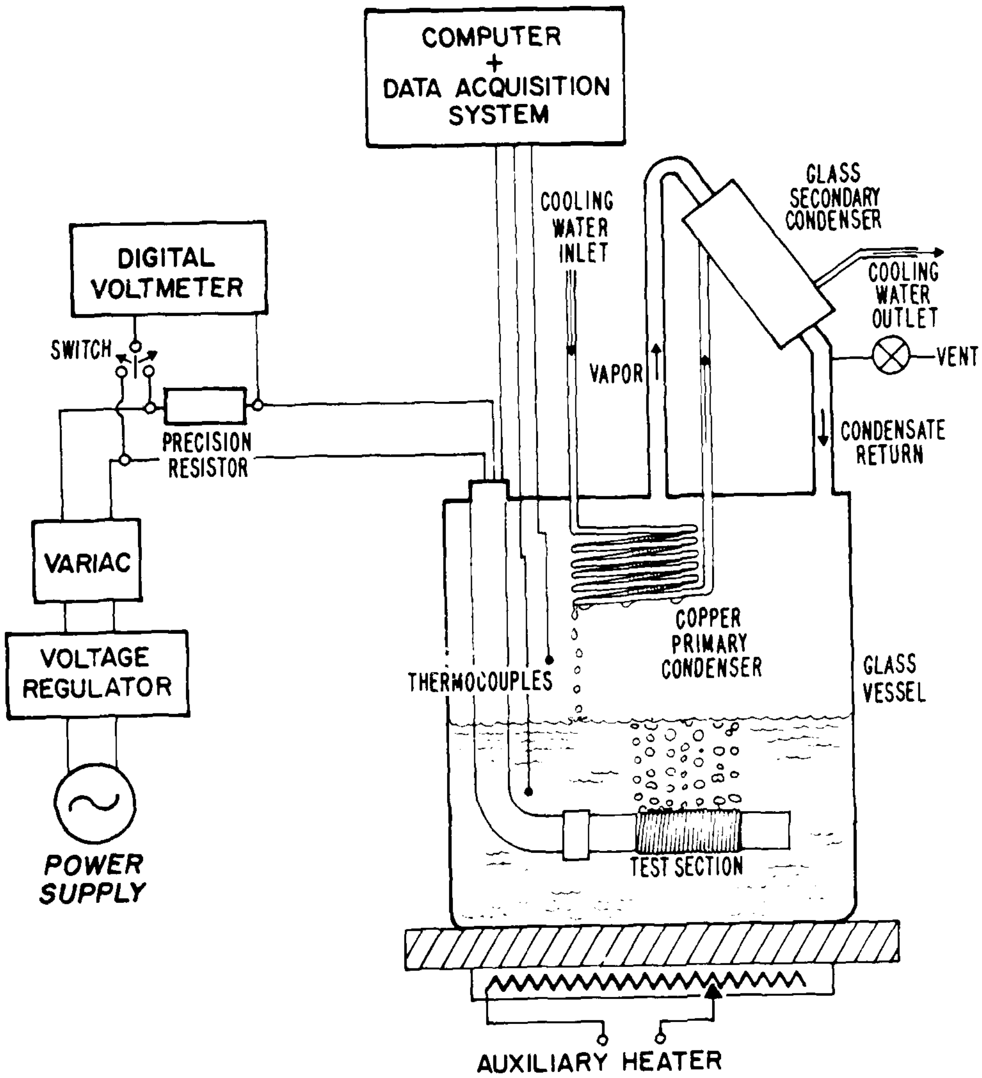

Figure 1. Schematic of Test Apparatus

Figure 1 shows a schematic of the test apparatus. The boiling surfaces consist of a cylindrical copper tube with a smooth interior surface and a commercially-available T-shaped finned surface manufactured by Wieland-Werke, AG. Each tube was conditioned for the insertion of thermocouples by drilling eight longitudinal holes as indicated in Figure 2 and introducing the respective copper-constantan thermocouples centered by means of capillary copper tubes. In the center of the tube a cartridge heater was placed to act as the heat source as shown in Figure 3.

Both ends of the test section were machined to remove the enhanced surface on the outside and to increase the interior diameter. This reduced the cross sectional area and thereby minimized the heat transfer from the ends of the test section. A Teflon plug was inserted into the free end for sealing and heat transfer reduction purposes. The other end of the test section was connected to a long copper extension tube through a 90-degree elbow, which allowed the test section to be placed in a horizontal position within a thick-walled Pyrex glass vessel. Since one of the requirements of the apparatus was to maintain the liquid at saturation temperature and atmospheric pressure, a plate heater and a vent tube were provided to satisfy these conditions. In addition avoiding excessive loss of vapor, a water-cooled copper coil was placed in the vapor space of the vessel, to act as a primary condenser. The vessel was covered with a

II. EXPERIMENTAL APPARATUS

A. REQUIREMENTS GOVERNING DESIGN

The purpose of this research was the observation of nucleate pool boiling from different Gewa-T surfaces. In order to get a more comprehensive knowledge about the physical mechanisms that govern this phenomenon, it is necessary to measure the parameters involved which affect the nucleate boiling process. The selected parameters to be obtained are the following:

1. Barometric pressure
2. Test section wall temperature
3. Fluid bulk temperature
4. Vapor temperature
5. Surface heat flux

B. TEST APPARATUS

The experimental apparatus designed by Lepere [Ref. 9], and modified by Hernandez [Ref. 8], with further minor modifications, was used during this thesis work. To gather data more conveniently at a faster rate, a Hewlett Packard model 9826 computer and an HP-3054A automatic data-acquisition system were added to the experimental facility. Lepere [Ref. 9] and Hernandez [Ref. 8] provide detailed descriptions of the apparatus; therefore, only a brief description is presented in this thesis.

to improve this performance through the use of further enhancing techniques.

There were four major objectives in this thesis study:

1. Take data to verify results of Hernandez [Ref. 8] on a standard 21.2 mm O.D. Gewa-T tube with and without shrouds.

2. Use wire wraps in the inter-fin area to improve nucleate-boiling heat-transfer performance from Gewa-T tubes.

3. Take data using the combination of best shroud (found in 1) and the best number of wire wraps (found in 2) on Gewa-T tubes.

4. Take data to study the effect of fin-tip spacing and fin density on the heat-transfer performance of Gewa-T tubes.

increase in the heat-transfer coefficient when compared to plain surface. The High Flux surface was most effective over a broad range of heat fluxes. The Thermoexcel-E surface showed similar gains in heat-transfer coefficient to that of the High Flux surface below 10 kW/m^2. The Gewa-T surface was not as effective as the other surfaces at low heat fluxes but performed comparably at high heat fluxes (near 100 kW/m^2).

Marto and Hernandez [Ref. 7] studied the boiling performance of a Gewa-T finned surface in Freon-113. They examined the influence of placing aluminum shrouds around the test tubes, and found that the thermal performance of these surfaces were dependent upon the liquid-vapor flow within the channels between neighboring T-shaped fins. They also examined the influence of the fins by progressively machining away the fin height to arrive at a smooth cylindrical surface. They reported that the addition of the "T-caps" to straight fins produced the most significant improvement in heat-transfer performance when compared to a smooth tube. The use of shrouds increased the heat-transfer coefficient at all practical heat fluxes, but decreased the heat-transfer coefficient at very high heat fluxes when compared to the unshrouded surface.

B. THESIS OBJECTIVES

Based on the foregoing discussion, the major motivation in this work was to further study the boiling heat-transfer performance of Gewa-T surfaces in R-113, and to attempt

development of generalized correlations or successful theoretical models.

Yilmaz, et al. [Ref. 5] compared the nucleate pool-boiling heat-transfer performance of three copper tubes: Linde High Flux, Wieland Gewa-T and Hitachi Thermoexcel-E. Their data taken with p-xylene showed that while all tubes performed better than smooth tubes, the best performance was with the Linde High Flux surface. The Thermoexcel-E tube performed better than the Gewa-T tube at low heat flux and both performed similarly at high heat flux. These data were taken under the fully-established boiling regime; thus, no information was provided on the hysteresis behavior or temperature overshoot of these surfaces.

Bergles and Chyu [Ref. 6] compared the nucleate pool-boiling heat-transfer performance of four different Linde High Flux tubes to a plain copper tube in distilled water and in R-113. They used three different methods of aging the test surface prior to collecting data. They reported significant temperature overhsoots of both plain and High Flux surfaces in R-113. They also reported that these overshoots were sensitive to aging, initial subcooling and rate of power increase.

Marto and Lepere [Ref. 1] obtained results on the heat-transfer performance of three heat-transfer surfaces: Linde High Flux, Thermoexcel-E, Gewa-T, and a plain copper surface in R-113 and FC-72. They reported a two to tenfold

Nishikawa and Ito [Ref. 4] recently discussed two methods to augment nucleate boiling: (1) use teflon-coated pits to reduce wettability of boiling liqud, and (2) manufacture surfaces with numerous re-entrant cavities that have the ability to trap vapor and keep the nucleation sites active. Due to the high wettability of low surface-tension fluids (such as fluorcarbons), the use of teflon-coated pits has not been successful. Thus, the surfaces with re-entrant-type cavities are the viable candidates for low surface-tension liquids.

Taking advantage of the concept of re-entrant cavities, a number of enhanced boiling surfaces have been invented during the last two decades. They include the Linde High Flux surface of Union Carbide, Gewa-T surface of Wieland and Thermoexcel-E surface of Hitachi. All of these surfaces are provided with re-entrant cavities through specialized manufacturing processes. These re-entrant cavities introduce yet more parameters to be desired.

In view of the inadequate understanding of the nucleate pool boiling process on smooth tubes, a thorough understanding of the boiling process on advanced surfaces may take many more years to come. Thus, present-day designers have very little information when considering advanced boiling surfaces. It is very important that researchers gather a vast amount of experimental data covering a wide range of heat-transfer fluids. Such extensive data will enable the

good indication of the complex nature of the nucleate boiling process. Chongrungreon and Sauer [Ref. 2] summarized nine(9) correlations for nucleate pool boiling of single-component fluids from smooth tubes. They stated the inadequacy of any of these correlations when compared with experimental data.

Nucleate pool boiling is strongly dependent on the surface structure; i.e., the heat flux increases with increase in stable nucleation site density. The size, shape and density of such nucleation sites are almost impossible to predict. However, the entire nucleate pool-boiling process is dependent on the performance at these nucleation sites. It is well known that the ability of these sites to develop and trap small vapor bubbles is the reason for improved performance in nucleate pool boiling in comparison with natural-convective boiling.

When a bubble is being generated at a surface nucleation site (or a pore), the wall superheat is given by the following approximate theoretical equation [Ref. 3]:

$$T_W - T_S = \frac{2\ \sigma\ T_S}{\rho_V\ h_{FG}\ R}$$

Thus, for a given fluid, the wall superheat can be decreased (boiler performance increases with decrease in wall superheat) with increase in pore radius. Of course, this radius must have a practical upper limit since too large a pore would not trap a vapor bubble.

I. INTRODUCTION

A. BACKGROUND

The energy crisis that ! ˉαan in 1974 has made a considerable impact on the design and operation of energy-conversion devices. Today, designers are challenged to design more and more efficient heat-transfer systems. Among the numerous areas that designers are faced with this challenge, boilers or evaporators used in many engineering applications demand considerable attention. The use of advanced boiling surfaces may lead to significant improvements in efficiency and/or significant savings in the initial cost and equipment sizes. Also, due to the ever-decreasing sizes of electronic components, the presence of very high heat fluxes has made the cooling of such components a challenging problem. Marto and Lepere [Ref. 1] stated that the use of advanced boiling surfaces may have potential promise in the field of electronic cooling. However, the lack of generalized performance data and useful theoretical models for such advanced surfaces have still kept designers away from the utmost use of these surfaces.

Due to a lack of complete understanding, the theoretical treatment of the nucleate pool-boiling process, even on a smooth surface, is extremely complex. The existence of a large number of semi-empirical formulas is probably a very

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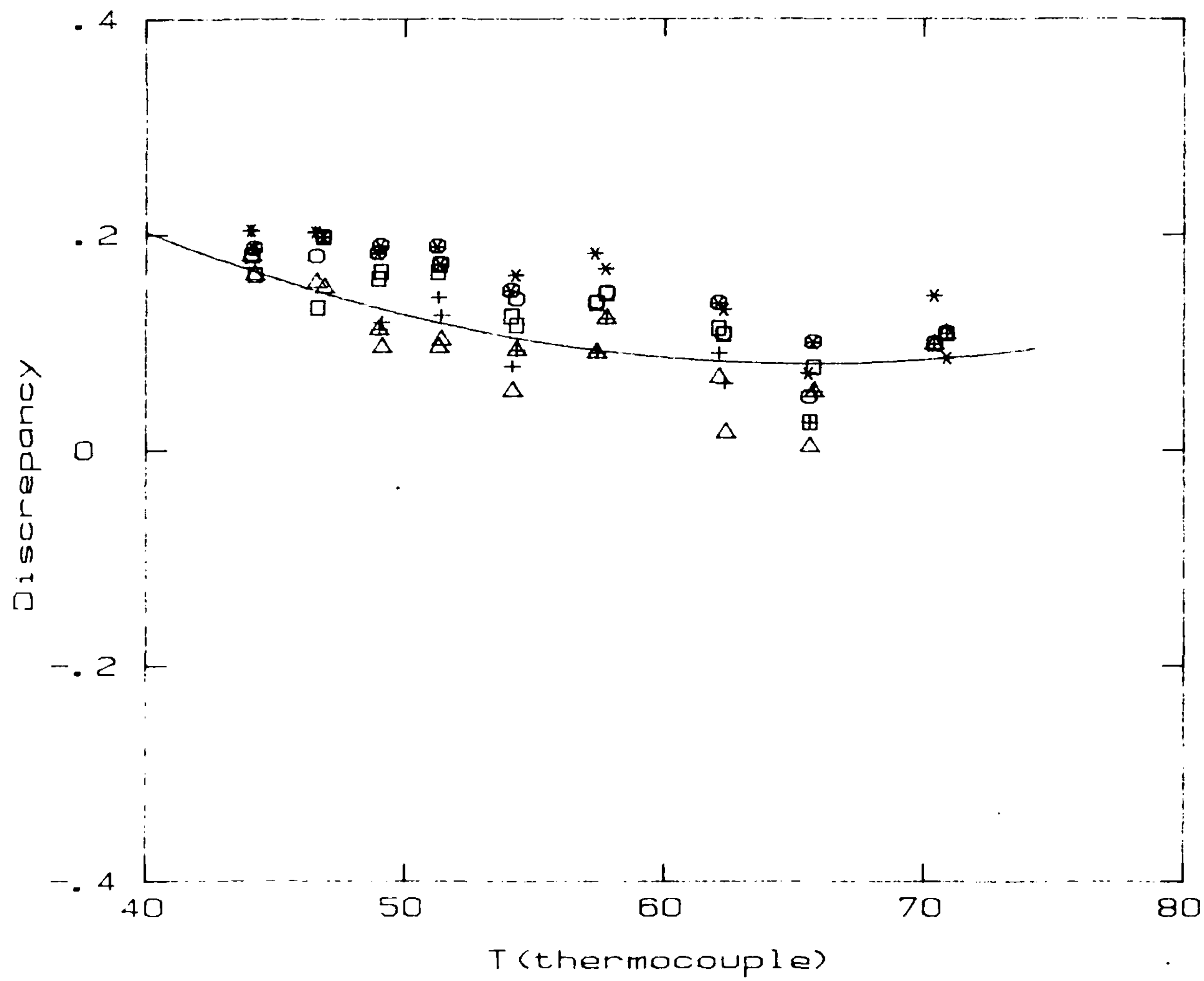


Figure 4. Thermocouple Calibration Curve

calibration line. Based on this fact, no calibration was performed for the other five tubes. Instead, the calibration curve generated for tubes 1 and 7 was assumed for all tubes.

III. EXPERIMENTAL PROCEDURES

A. NORMAL OPERATION

The Procedure B employed by Marto and Hernandez [Ref. 7] has been demonstrated to be the most adequate to carry out the boiling runs. Because of the large influence of surface past history on boiling incipience, this procedure was established to allow repeatability and permit comparison of the obtained data for different enhancements.

The procedure is as follows:

1. Once the test section has been immersed in the pool, it was subjected to one hour of pre-boiling by setting the cartridge heater to give a heat flux of about 30 kW/m^2. The plate heater was adjusted to the maximum temperature for this one-hour period. This initial vigorous boiling served to degas the liquid and to force the noncondensable gases out to the atmosphere.

2. After the initial aging process, both test-section heater and plate heater were secured, allowing cooling of the liquid and the test section for about 30 minutes.

3. The plate-heater voltage was then adjusted in order to maintain the liquid at saturated conditions and the cartridge heater was turned on to the first setting of 12 volts.

4. For all the consecutive settings during both increasing and decreasing heat fluxes, the boiling was allowed to stabilize for five minutes at each power setting.

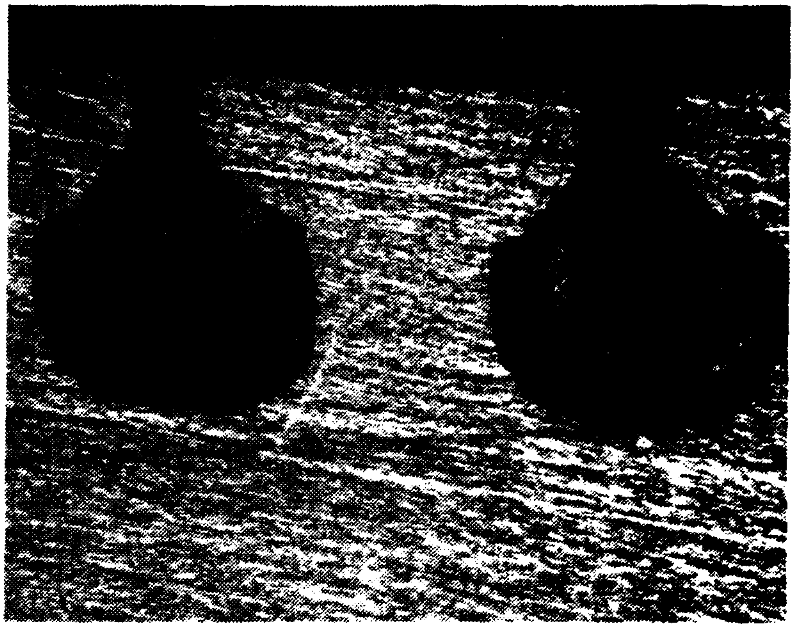
5. At a given power setting, the following data were recorded: heater voltage, precision-resistor voltage, vapor temperature, liquid bulk temperature and wall temperatures of the test section.

B. GEWA-T SURFACES

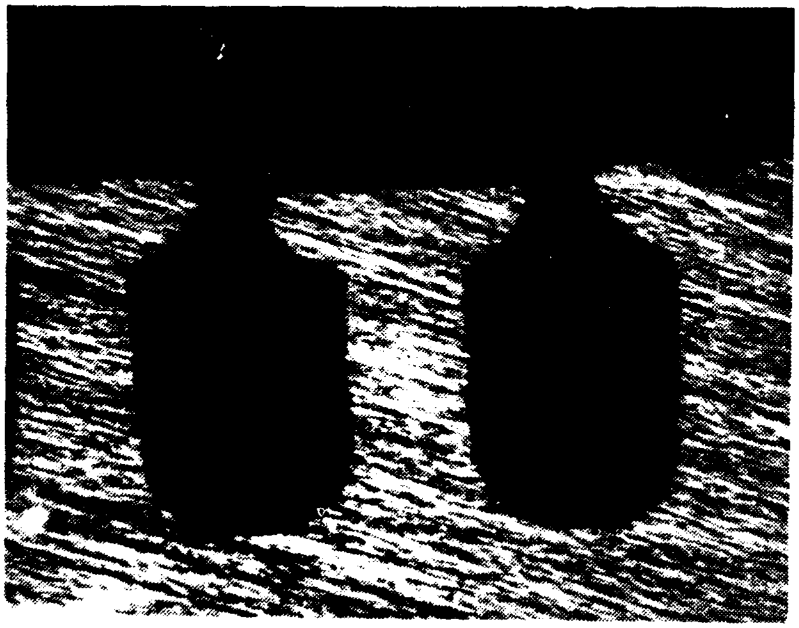
Three groups of surfaces were tested. The first surface was the same as used by Hernandez [Ref. 8] with a fin density of 0.75 fins/mm, an outside diameter of 21.2 mm, and an inter-fin gap of 0.25 mm. The second group of three tubes had 0.75 fins/mm, an outside diameter of 23.1 mm and inter-fin gaps of 0.15, 0.25 and 0.35, and the third group of three tubes had 1.02 fins/mm and all other specifications were the same as the second group.

The geometry of the enhancement of these seven Gewa-T surfaces was observed by making a longitudinal cut of the tubes and photographing them with 50X magnification (Figure 5). As can be seen, the tubes with 0.75 fins/mm had an approximately circular cavity shape, while the 1.02 fins/mm tubes had a somewhat elongated cavity shape. This difference in shape was probably due to the technique used in construction to obtain the various specifications. All the T-shaped fins had a fin height of approximately 0.95 mm.

The exact determination of diameters at the base of the fins was carried out from the magnified photographs of the cavities. This diameter was determined by subtracting twice the fin height from the measured outside diameter:



(a) Gewa-T Tube Number 2



(b) Gewa-T Tube Number 5

Figure 5. Optical Micrograph of Gewa-T Fin Profile, 50X

$$D_2 = O.D. - 2H_F ,$$

resulting in the dimensions presented in Table 1.

TABLE 1

Dimensions of Gewa-T Finned Tubes

TUBE NO.	FINS/ mm	FIN TIP GAP (mm)	FIN HEIGHT (mm)	O.D. (mm)	D_2 (mm)
1	0.75	0.25	0.865	21.2	19.47
2	0.75	0.15	0.980	24.99	23.05
3	0.75	0.25	0.960	24.99	23.07
4	0.75	0.35	0.970	24.99	23.05
5	1.02	0.15	1.050	25.30	23.20
6	1.02	0.25	1.050	25.30	23.20
7	1.02	0.35	1.125	25.30	23.05

For simplicity, the diameter at the base (D_2) was assumed to be 19.5 mm for Tube 1 and 23.1 mm for all other tubes.

C. HEAT-FLUX CALCULATION

According to earlier description about test-section construction, the cartridge heater acts as the heat source, inserted into the tube, and variations in heat-transfer rate are made through different power settings with a variac.

$$Q_H = VI \tag{1}$$

where:

Q_H = heat-transfer rate from the cartridge heater (W)

V = voltage across the cartridge heater element (volts)

I = current through the heater element (amps)

This circuit is connected in series with a precision resistor, which provides the means to compute the electrical current by the measurement of the voltage drop across the resistor:

$$I = V_{RES}/R \tag{2}$$

where:

V_{RES} = voltage drop across the precision resistor (volts)

R = resistance of the precision resistor (ohms)

= 2.031 ohms

Both ends of the Gewa-T tube were machined to provide a smooth surface and a reduced wall thickness in order to minimize longitudinal heat conduction and therefore the heat losses from the unenhanced ends. The geometry of these ends is shown in Figure 6, where:

b = base of the T-shaped fins

T_b = temperature at the base of T-shaped fins (°C)

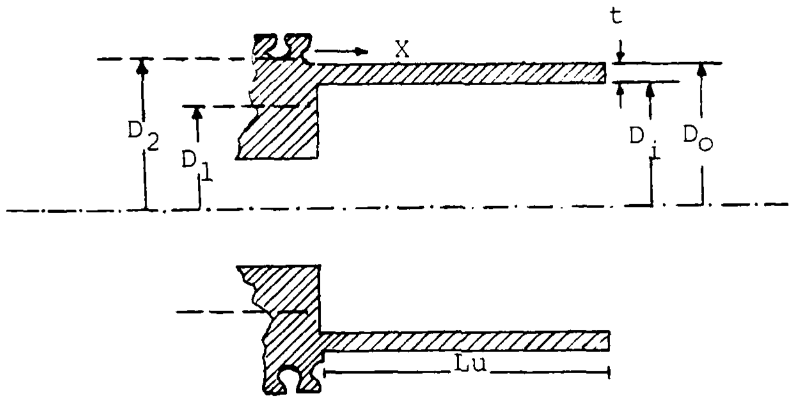


Figure 6. Geometry of Unenhanced Ends

For this cylindrical geometry:

$$A_C = \frac{\pi}{4}(D_o^2 - D_i^2) \tag{3}$$

$$A_S = P \times L_c \tag{4}$$

where:

A_C = cross-sectional area of tube wall (m^2)

D_o = tube outside diameter (m)

D_i = tube inside diameter (m)

A_S = tube outside surface area (m^2)

P = tube outside wall perimeter (m)

It was assumed that the temperature at the base of the thin-walled fin was equal to the temperature at the base of the T-shaped fins. Therefore:

$$\theta(x) = T(x) - T_{SAT} \tag{5}$$

In general,

$$\frac{d^2\theta}{dx^2} - m^2\theta = 0 \tag{6}$$

where:

$$m^2 = \frac{\bar{h}\,P}{kA_C} \tag{7}$$

Therefore

$$Q_F = Q_B = -kA_C \left.\frac{dT}{dx}\right|_{x=0} = -kA_C \left.\frac{d\theta}{dx}\right|_{x=0} \tag{8}$$

Now, assuming negligible convective heat loss from the thin tube tip:

$$\left.\frac{d\theta}{dx}\right|_{x=L_U} = 0 \tag{9}$$

and

$$\frac{\theta}{\theta_B} = \frac{\text{Cosh } m(L-x)}{\text{Cosh } mL} \tag{10}$$

For this case

$$Q_F = \sqrt{\bar{h}\,P\,k_C\,A_C}\;\theta_B \text{ Tanh } mL_U \tag{11}$$

$$\theta_B = \theta(o) = T_B - T_{SAT} \qquad (12)$$

where:

Q_F = heat-transfer rate through unenhanced end (W)

T_{SAT} = fluid saturation temperature (°C)

k_C = thermal conductivity of the copper (W/m.K)

$\bar{h}$ = average heat-transfer coefficient (W/m^2.K)

Assuming the wall temperature is the same as the average of the eight calibrated wall temperature thermocouple measurements, less temperature drop due to radial conduction from thermocouple position to the base of the T-shaped fin,

$$T_B = T_{AVG} - Q_H \frac{\ln(D_2/D_1)}{2 \pi L k_C} \qquad (13)$$

$$T_{AVG} = \left(\sum_{n=1}^{8} T_n\right)/8.0 \qquad (14)$$

where:

D_1 = diameter at the position of thermocouples (m)

T_{AVG} = average wall temperature (°C)

For the geometry shown in Figure 6:

$$L_C = L_U + \frac{t}{2} \qquad (15)$$

$$t = D_o - D_i \qquad (16)$$

where:

L_C = corrected length (m)

L_U = length of unenhanced ends (m)

t = tube thickness (m)

The average difference between the wall temperature and the saturation temperature may now be determined from the following equation:

$$\bar{\theta} = \frac{1}{L_C} \int_0^{L_C} \frac{\text{Cosh } m(L_C - x)}{\text{Cosh } mL_C} dx \qquad (17)$$

$$\bar{\theta} = \frac{\theta_B}{mL_C} \text{ Tanh } mL_C \qquad (18)$$

$$\bar{\theta} = \bar{T}_W - T_{SAT} \qquad (19)$$

where:

$(T_W - T_{SAT})$ = average difference between surface wall temperature and fluid saturation temperature (K)

Since the unenhanced ends are very long and the thickness is very small, only very little heat transfer would take place at these ends. Thus, these ends undergo free convection,

for which the Churchill and Chu correlation as stated in [Ref. 10] for the average Nusselt number was assumed:

$$Nu_{D_o} = \left\{ 0.60 + \frac{0.387\ \overline{Ra}_{D_o}^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}} \right\}^2 \tag{20}$$

$$10^{-5} < \overline{Ra}_{D_o} < 10^{12}$$

The average Nusselt number is:

$$\overline{Nu}_{D_o} = \frac{\bar{h}\ D_o}{k} \tag{21}$$

where:

k = thermal conductivity of Freon-113 (W/m.K)

Solving for h:

$$\bar{h} = \frac{k}{D_o} \left\{ 0.60 + \frac{0.387\ \overline{Ra}_{D_o}^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}} \right\}^2 \tag{22}$$

$$Pr = \frac{\nu}{\alpha} \tag{23}$$

$$\overline{Ra}_{D_o} = \frac{g\ \beta\ (T_W - T_{SAT})\ D_o^2}{\nu\ \alpha} \tag{24}$$

$$\beta = -\frac{1}{\rho}\frac{\Delta\rho}{\Delta T} \tag{25}$$

where:

Pr = Prandtl number

ν = kinematic viscosity (m^2/s)

α = thermal diffusivity (m^2/s)

$\overline{Ra}_{D_o}$ = average Rayleigh number

g = gravitational acceleration (m/s^2)

β = volumetric thermal expansion coefficient ($1/K$)

ρ = density of freon-113 (kg/m^3)

Now, substitution of Equations 7, 18 and 24 into Equation 22 results in:

$$\bar{h} = \frac{k}{D_o}\left\{0.60 + \frac{0.387\left[\dfrac{g\beta D_o^3 \theta_B \mathrm{Tanh}\left\{\left(\dfrac{\bar{h}P}{kA_C}\right)^{1/2} L_C\right\}}{\nu\alpha L_C\left(\dfrac{\bar{h}P}{kA_C}\right)^{1/2}}\right]^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}}\right\}^2 \qquad (26)$$

Equation 26 is solved for $\bar{h}$ by iterative technique within a range of precision of 0.001. Knowing the value of natural-convective heat-transfer coefficient at unenhanced ends, the total heat-loss rate is calculated from Equation 11, and the heat-transfer rate through the enhanced Gewa-T surface is obtained by subtracting the total heat-loss rate from the total heat-transfer rate:

$$Q_{LOSS} = 2Q_F \qquad (27)$$

$$Q = Q_H - 2Q_F \tag{28}$$

Finally, the heat flux from the Gewa-T surface at the base of the T-shaped fins is:

$$q = Q/A_B \tag{29}$$

$$A_B = \pi D_2 L \tag{30}$$

diameter. For this purpose, a semi-empirical correlation developed by Cornwell et al. [Ref. 11] is used:

$$Nu_{TB} = C_{TB}\ Re_{TB}^{2/3}$$

$$\frac{h_{TB}\ D}{k_F} = C_{TB}\left(\frac{qD}{h_{FG}\ \mu_F}\right)^{2/3}$$

$$\frac{\rho_F\ D\ V\ A}{\mu_F\ A} = \frac{\dot{m}\ D}{\mu_F A} = \frac{GD}{\mu_F}, \qquad G = q/h_{FG}$$

$$h_{TB} = \frac{C_{TB}\ k_F}{D}\left(\frac{h_{TB}\ \Delta T\ D}{h_{FG}\ \mu_F}\right)^{2/3}$$

$$h_{TB}^{1/3} = \frac{C_{TB}\ k_F}{D}\left(\frac{\Delta T\ \ D}{h_{FG}\ \mu_F}\right)^{2/3}$$

$$h_{TB} = \left(\frac{C_{TB}\ k_F}{D}\right)^3 \left(\frac{\Delta T\ D}{h_{FG}\ \mu_F}\right)^2$$

$$h_{TB} = (C_{TB}\ k_F)^3 \left(\frac{\Delta T}{h_{FG}\ \mu_F}\right)^2 \frac{1}{D}$$

Using this relationship, Hernandez's data were revised and a least-squares-fit curve was generated. This curve is also plotted in Figure 12.

H. BOILING PERFORMANCE OF GEWA-T TUBE WITH SHROUDS

The same four shrouds tested by Hernandez were used during this investigation. These shrouds had the following geometry:

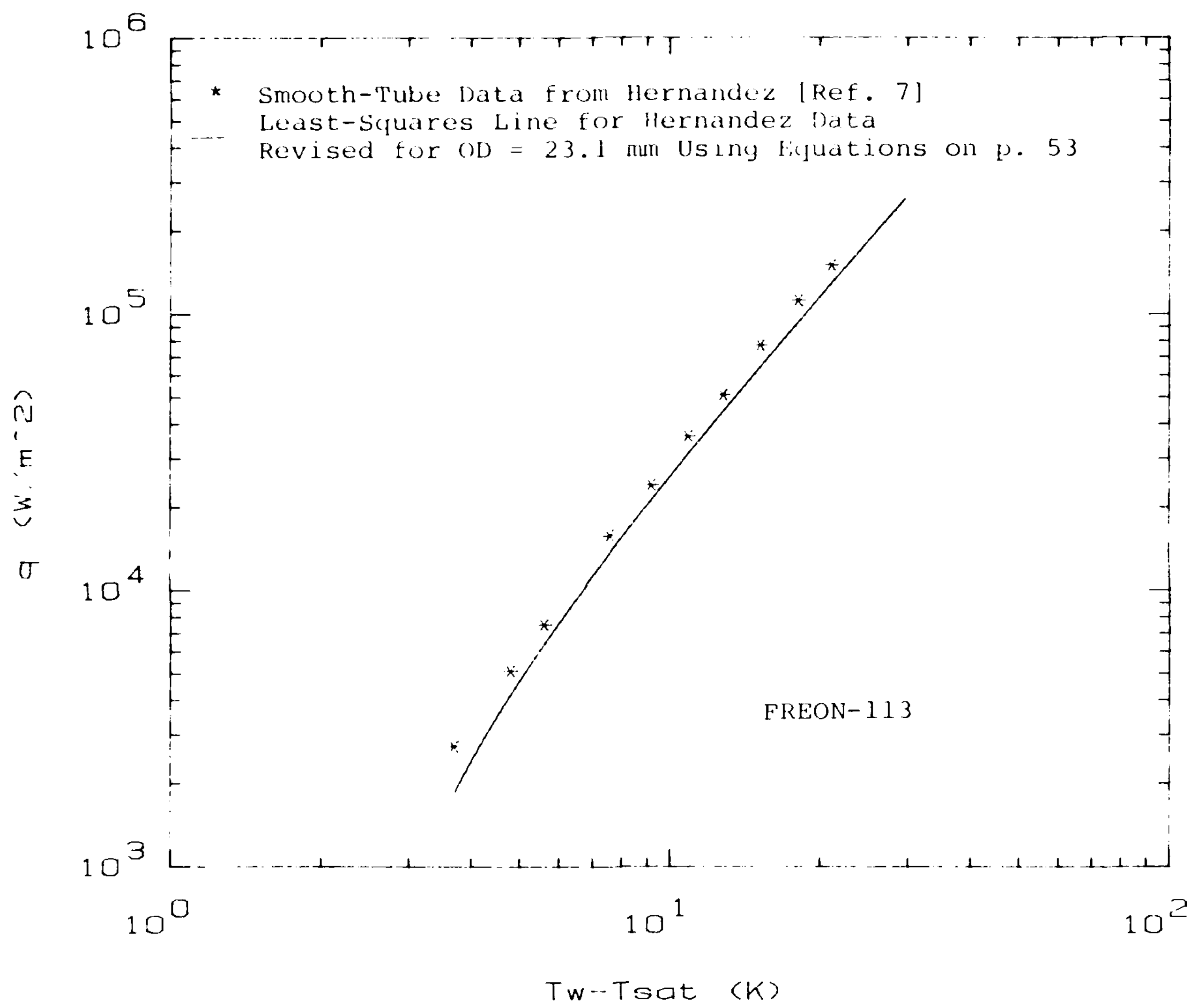


Figure 12. Extrapolation to Smooth Tube (23.1 mm OD) from Smooth-Tube Data of Hernandez (19.61 mm OD)

agree to within 0.2 percent at a heat flux of 30,000 to 100,000 W/m^2. These data runs show considerably good repeatability of this experiment. (Similar agreements were also found for all other tubes.)

Also, these data show a considerable enhancement of Gewa-T tube performance over the "Smooth" tube data of Hernandez. For example, at $\Delta T = 3.1$ K, the Gewa-T tube gives a 4.65 times larger heat flux than the smooth tube.

As also discussed by Lepere [Ref. 9] and Hernandez [Ref. 8] this enhancement in boiling performance is mainly attributed to the existence of re-entrant cavities in the Gewa-T tube.

G. BOILING PERFORMANCE OF SMOOTH TUBE

To compare the performance of Gewa-T tubes to smooth-tube performance, data of Hernandez were chosen. It is important to point out that his smooth tube did not consist of a commercially-available surface. Instead, Hernandez obtained his "Smooth" surface by machining fins off of a Gewa-T tube. He did not remove fins completely; the remaining portions on the tube measured about 0.1 mm.

Figure 12 shows Hernandez's data (heat flux as a function of wall superheat) for only decreasing heat fluxes. A least-squares-fit curve generated for these data points will be used in future figures representing data on tube 1. Since tubes 2 through 7 have a larger diameter, it was necessary to revise Hernandez's data to include the difference in

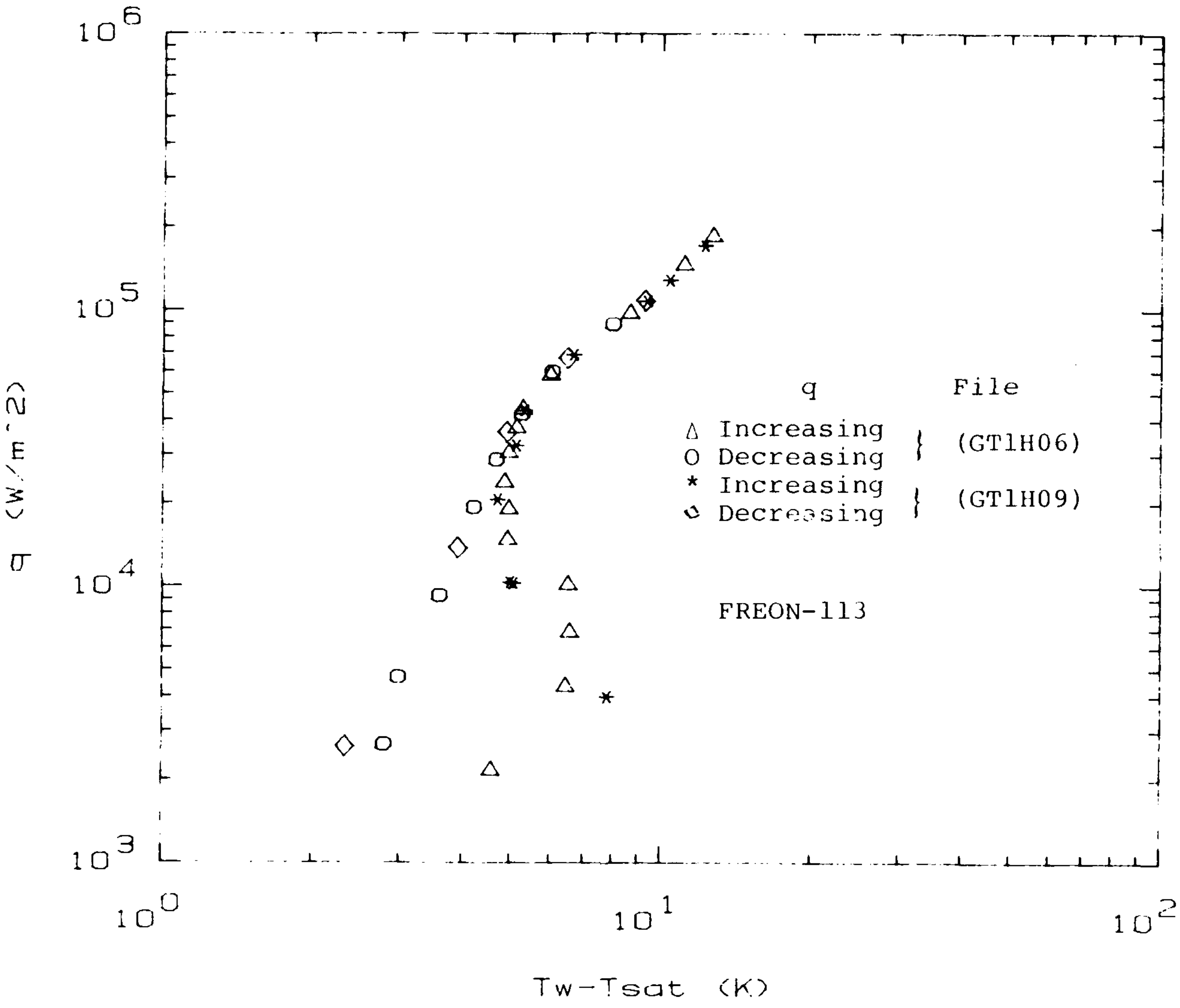


Figure 11. Reproducibility Test: Plain Gewa-T Tube 1

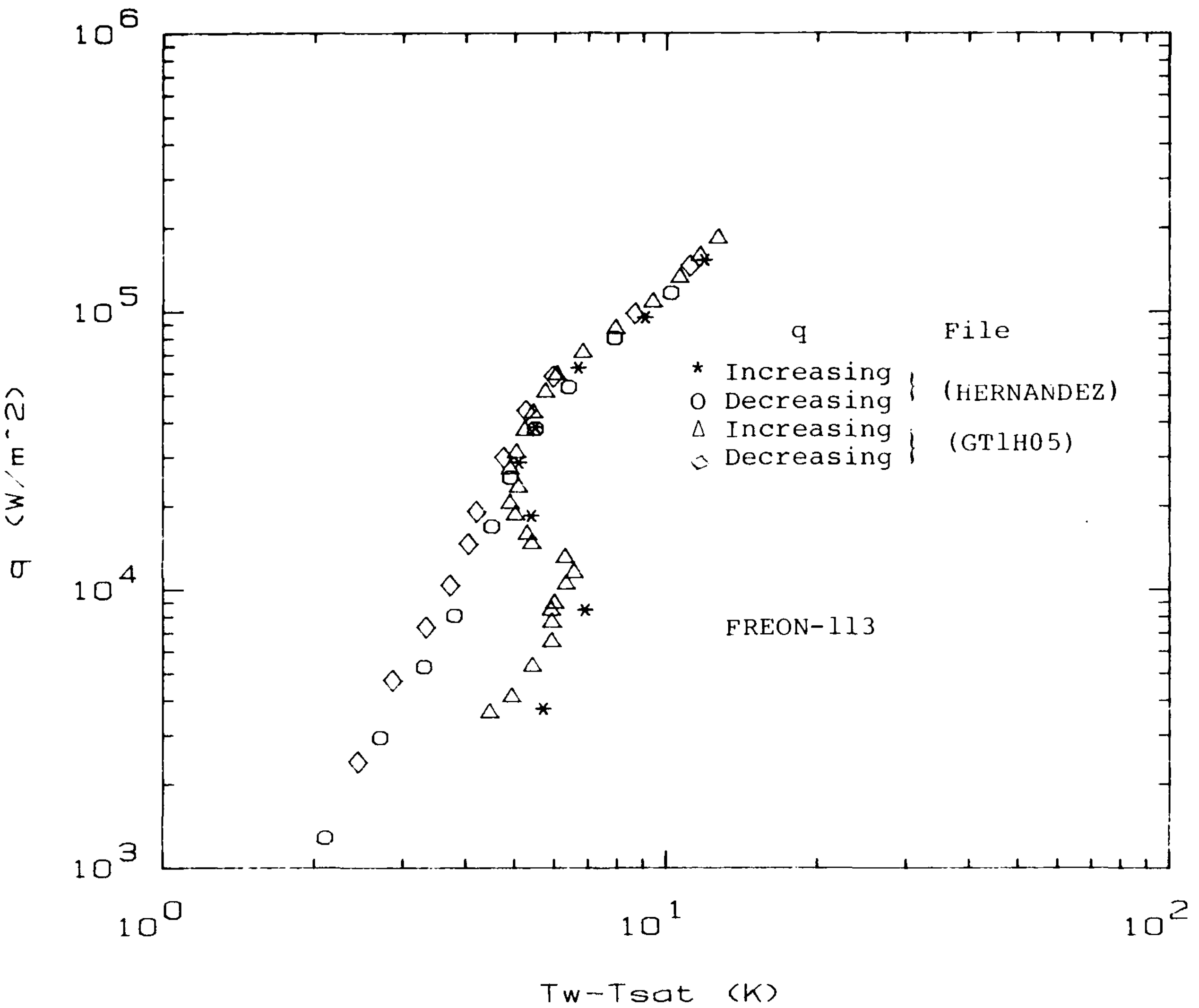


Figure 10. Reproducibility Test: Hernandez Run No. 3 Versus Present Work (GT1H05)

E. REPEATABILITY WITH PREVIOUS DATA

Following modifications made, especially to the data gathering means (i.e., automatic data-acquisition system and newly-written computer programs), it was necessary to take data for tube 1 (this is the same tube tested by Hernandez) to check the present data with Hernandez data under similar conditions. Even though a total of 5 different runs were made during this stage through debugging of programs and system familiarization, only run number 5 will be presented in this thesis with a comparison to Hernandez data.

Figure 10 shows the comparison between the present data and the data of Hernandez. A very good agreement is evident as the two sets disagree by less than 2 percent for the nucleate boiling regime. However, considerable disagreement exists in the region of increasing heat flux before total nucleate boiling. This is believed to be due to the rather random behavior of the boiling process during this regime as it will be presented in the next section, even two consecutive runs for a given tube would not give the same results during the unstable boiling regime.

F. REPEATABILITY STUDY OF PRESENT WORK

Figure 11 shows data from two different runs completed on two different days. As discussed earlier, a considerable disagreement can be seen before the tube undergoes complete nucleate-boiling process. However, for the complete nucleate-boiling process, the data from different data runs

confirmed the validity of the mathematical model assumed for data reduction. However, a small rate of bubble formation was evident at the tip of the unenhanced fin end; these bubbles were in fact generated at the base of the fin (i.e., between the Teflon plug and the inner surface of the fin).

The vapor bubbles generated in the inter-fin cavities are carried up into the channels around the tube due to buoyancy forces. As these bubbles travel up, they activate more and more nucleation sites as well as they help for the removal of larger bubbles from the channels. This phenomenon is generally referred to as the "Chimney Effect."

If the heat flux is too large (see Figure 7), the rate of vapor bubble generation may be too large that the interfin channels could be full of a vapor blanket that may disrupt the flow of liquid to the hot metal surface where active boiling takes places. At this point, further increase in heat flux may result in excessive wall temperatures. Under these conditions, vapor bubbles were seen ejecting radially outward from the test surface. Line D-E represents the boiling regime just described. It is clear that the slope (i.e., heat-transfer coefficient) at E is considerably smaller than that at point D. This decrease in heat-transfer coefficient is assumed to be due to the existence of an insulating vapor blanket.

When heat flux is gradually decreased, the curve follows a different path (D-F) as shown in Figure 7. This is due to the existence of stable nucleation sites that will remain active for a wide range of heat fluxes.

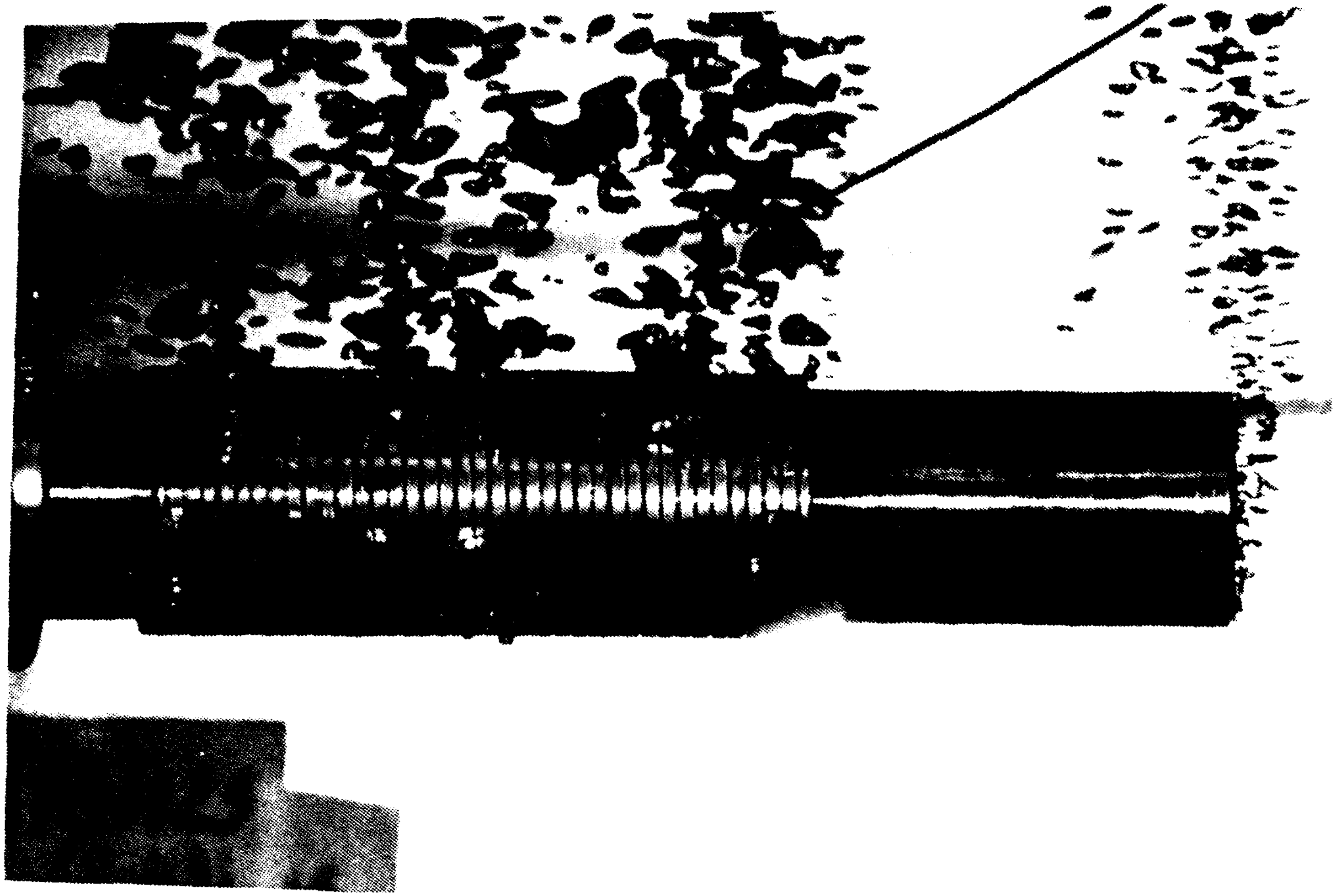


Figure 9. Photograph of Gewa-T Surface at Low Heat Flux

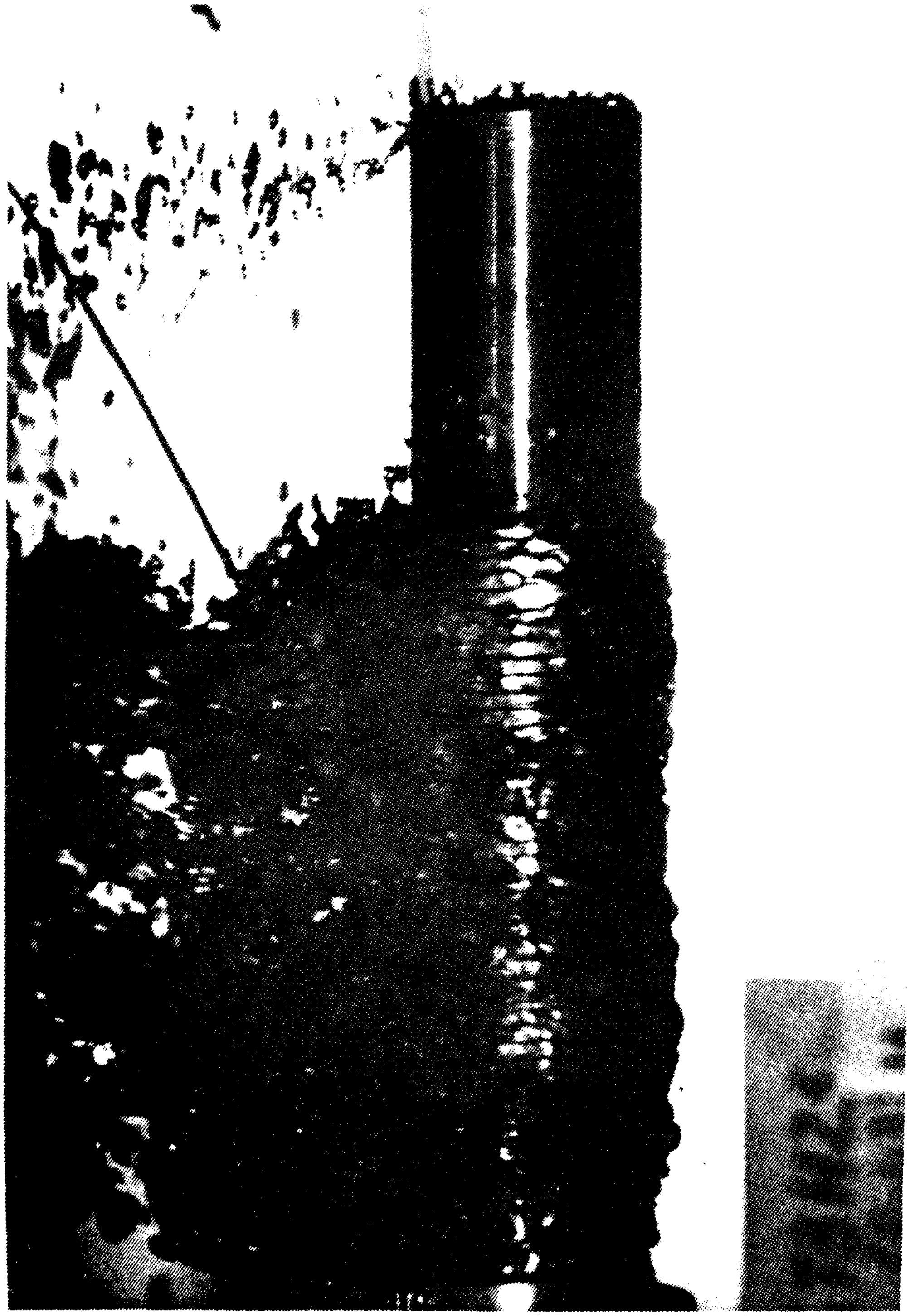
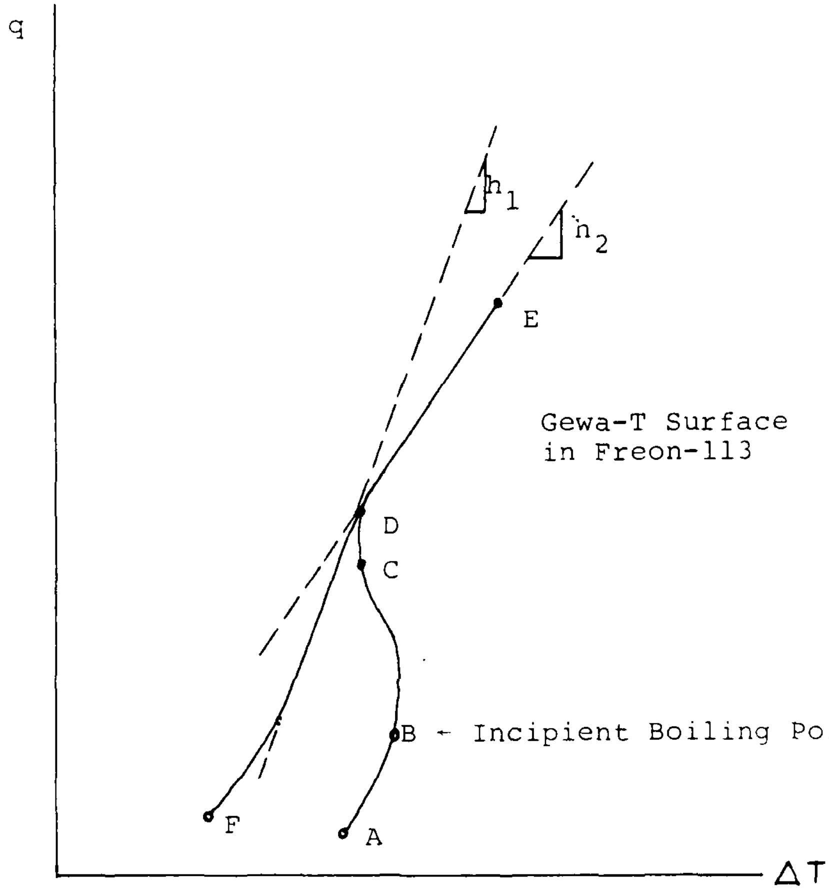


Figure 8. Photograph of Gewa-T Surface at High Heat Flux

From point A to point B, a continuous increase in heat flux, accompanied by an increase in wall superheat (T_W - T_{SAT}) is observed when heat flux is increased. This region corresponds to natural-convection process, as also verified during experimental runs with the absence of bubbles along both enhanced surface and unenhanced ends.

From point B to point C, a drastic reduction in wall superheat was observed while the heat flux continuously increased. This region was somewhat unstable as more and more nucleation sites became active. This region is known as the mixed boiling region, where transition from natural-convective boiling to nucleate pool boiling takes place. At these heat fluxes, visual observations revealed that the bubble formation was quite random. While some portions of the tube showed bubbles, other portions showed no bubble formation. The wall superheat continued to decrease until all nucleation sites became active, as indicated by point D.

Once all available nucleation sites have been activated, the wall superheat starts to increase with increasing heat flux as shown by region D to E. When boiling in this region, the entire tube showed very high density of bubble formation. At this point, it is important to point out that during all experimental runs, for the entire range of power settings (7.5 to 485 W), little or no nucleate pool boiling on the unenhanced surface of the ends was observed (see Figures 8 and 9) for the different test sections tested. This observation



A-B: Natural Convection
B-C: Mixed Boiling (Activation of Nucleation Sites)
C-D: Active Boiling
D-E: High Flux Nucleate Boiling
E-F: Stable Nucleate Boiling

Figure 7. Typical Nucleate Pool-Boiling Curve for a Gewa-T Surface

runs were made. All of these runs showed only a negligible variation of the wall thermocouple temperatures; maximum variation was ±0.25 K. These runs clearly showed that the heater element indeed provided a uniform heat flux along the circumferential direction. No detailed data are presented in this thesis for the tube in the vertical position. All other results presented herein are for the tubes in the horizontal position.

C. LONGITUDINAL UNIFORMITY OF HEAT FLUX ALONG THE TEST SECTION

During most of the data runs with the tubes in the horizontal position, somewhat unexplainable phenomena were observed. As the heat flux was gradually increased, bubble formation began simultaneously at the two ends, while the center portion of the tube underwent only natural convection. Approximately 40 percent of maximum heat flux was necessary to obtain nucleate boiling from the entire tube. Since all the wall thermocouples are located at midway along the tube, the average wall temperature values provided in all data runs may be in slight error.

D. PLOT ANALYSIS FOR PLAIN SURFACE

Figure 7 represents a typical curve for nucleate pool-boiling process of Gewa-T surface in Freon-113. The observed behavior of this process is analyzed in this section by studying the different sections of the boiling curve.

separate file. This plot data file could easily be recognized by the "P" added at the beginning of the raw data file name. In this manner, the plot data file corresponding to the above raw data file has the name PGT5H28.

A total of thirty different data runs were carried out during this investigation. These runs break down in the following manner:

Run Number	Tube Number	Test Condition
1 - 3	1	Plain - Vertical position
4 - 10	1	Plain - Horizontal
11 - 21	1	With shrouds and/or Wires
22 - 23	2	Plain - Horizontal
24 - 25	3	Plain - Horizontal
26	4	Plain - Horizontal
27 - 28	5	Plain - Horizontal
29	6	Plain - Horizontal
30	7	Plain - Horizontal

B. CIRCUMFERENTIAL UNIFORMITY OF HEAT FLUX AROUND HEATER ELEMENT

During the early stages of this investigation, it was thought necessary to test if the heater element would provide a non-uniform heat flux along the circumferential direction. Such a non-uniform heat-flux variation was assumed possible based on the heater-element construction. Therefore, tube 1 was immersed vertically in the R-113 liquid pool and three

IV. RESULTS AND DISCUSSION

A. DESIGNATION OF DATA FILES

A total of seven Gewa-T tubes were tested during this investigation. As can be seen from Table 1, the primary geometric variables were: fin-tip gap, fin density and outside diameter of tubes including fins.

The general designation for data files is:

GTNPnn

where:

- GT indicates Gewa-T surface
- N specifies the number of the tube tested (Table 1)
- P indicates the position of the test section (i.e., H for horizontal and V for vertical), and
- nn indicates the corresponding cumulative run number.

For example, GT5H28 specifies tube number five tested in the horizontal position during run number 28. From Table 1, the corresponding specifications for this test section are: 25.3 mm outside diameter, 23.1 mm diameter at the base of the fins, 1.02 fins/mm and inter-fin gap of 0.15 mm. While the file name just presented contain data in the very raw form, the processed data (i.e., q versus ΔT) were stored in a

Shroud Number	Upper Opening	Lower Opening
1	60°	60°
2	60°	30°
3	30°	30°
4	60°	8.5°

All of these shrouds had been designed for tube 1; thus, no shrouds were used for tubes 2 through 7.

Figures 13, 14, 15, and 16 show data on tube 1 with shroud numbers 1, 2, 3 and 4, respectively. As also reported by Hernandez [Ref. 8], all the shrouds improved boiling performance at low heat fluxes compared to plain Gewa-T tube. Table 2 lists performance calculations for tube 1 with various shrouds. It can be seen that the 60° × 8.5° shroud (number 4) produced the maximum improvement of 253 percent in the boiling coefficient over smooth tube at a heat flux of 10,000 W/m^2. At this heat flux, the 30° × 30° shroud resulted in a 165 percent improvement in the boiling heat-transfer coefficient. At a heat flux of 50,000 W/m^2, all shrouds resulted in about 190 percent improvement, except the 30° × 30° shroud which resulted in a 180 percent improvement.

At very high heat fluxes, for example at $q = 100,000$ W/m^2, the 60° × 60° shroud showed a maximum performance increase of 106 percent; the 60° × 30° shroud resulted in a 49 percent improvement; the 60° × 8.5° shroud resulted in a 18 percent increase; and the 30° × 30° shroud resulted in an 13 percent

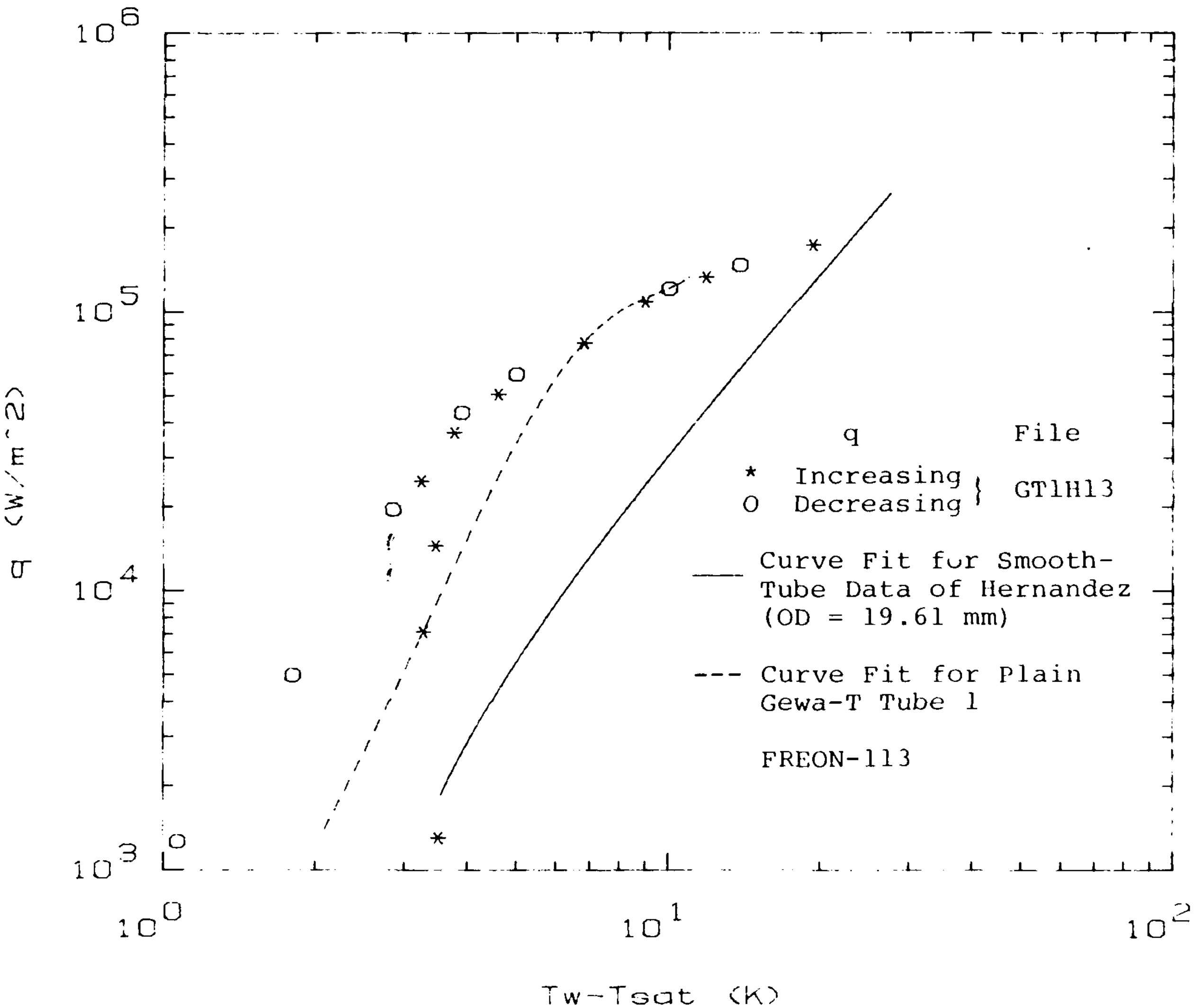


Figure 13. Effect of 60° × 60° Shroud on Boiling Performance of Plain Gewa-T Tube 1

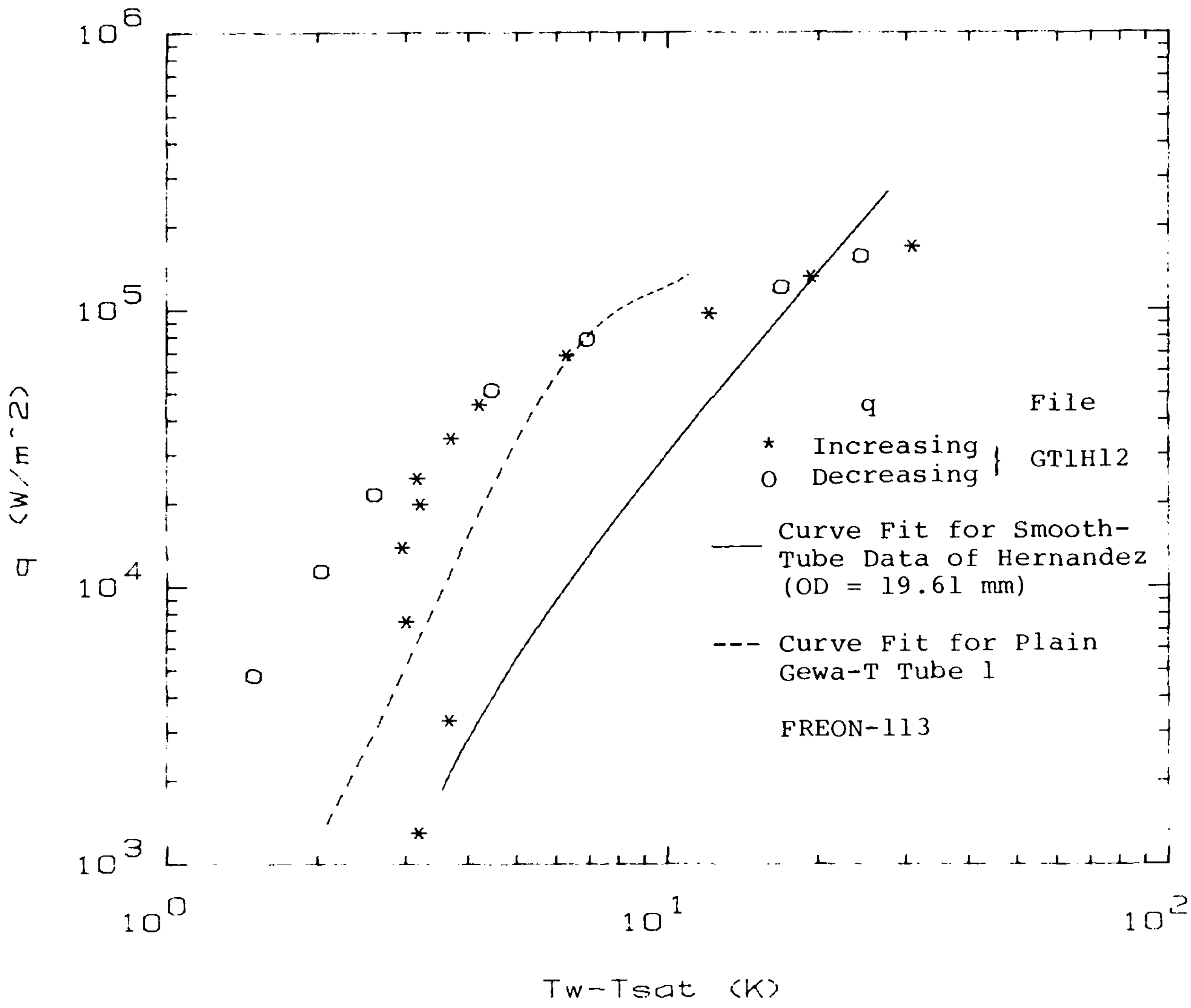


Figure 14. Effect of 60° × 30° Shroud on Boiling Performance of Plain Gewa-T Tube 1

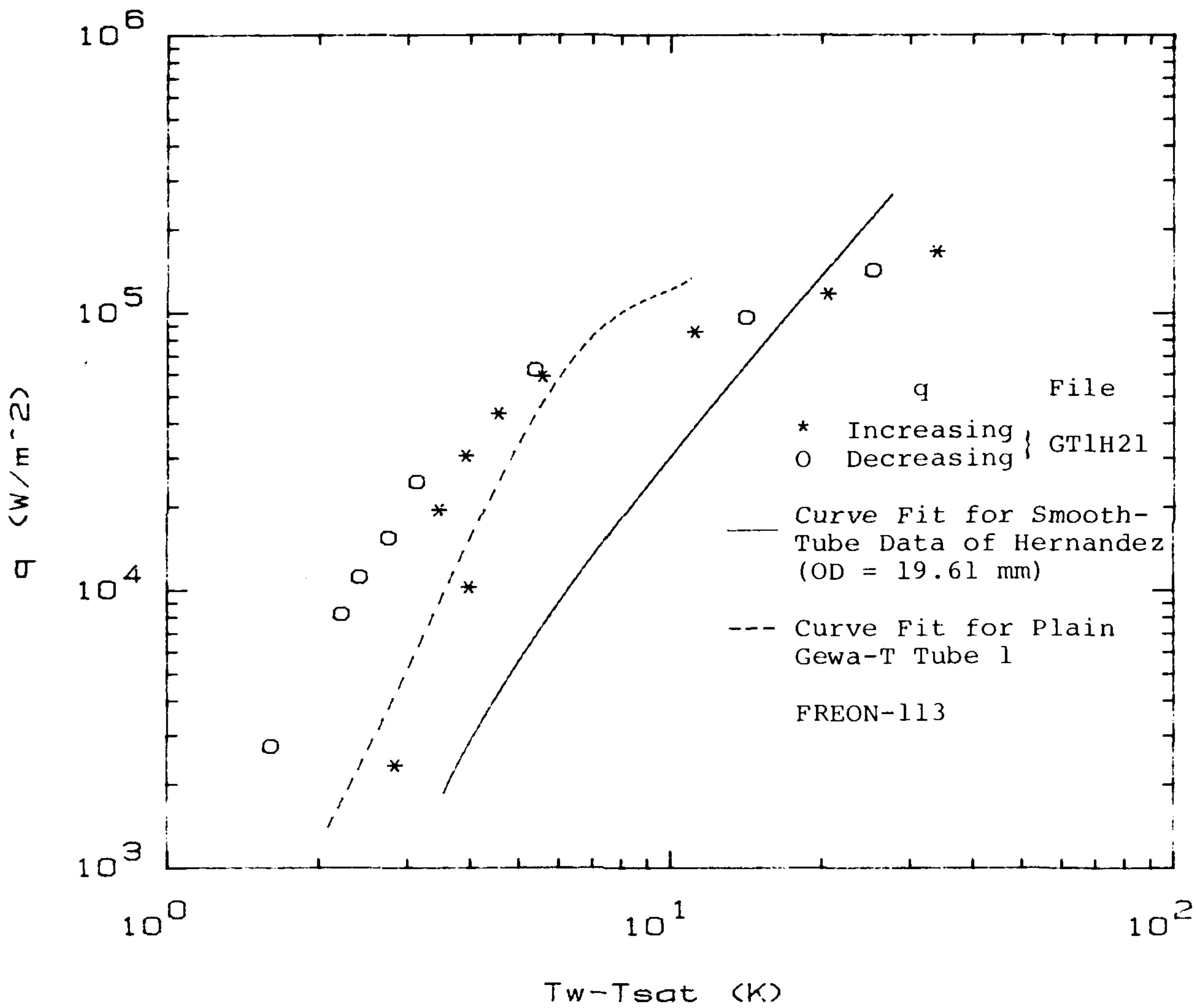


Figure 15. Effect of 30° × 30° Shroud on Boiling Performance of Plain Gewa-T Tube 1

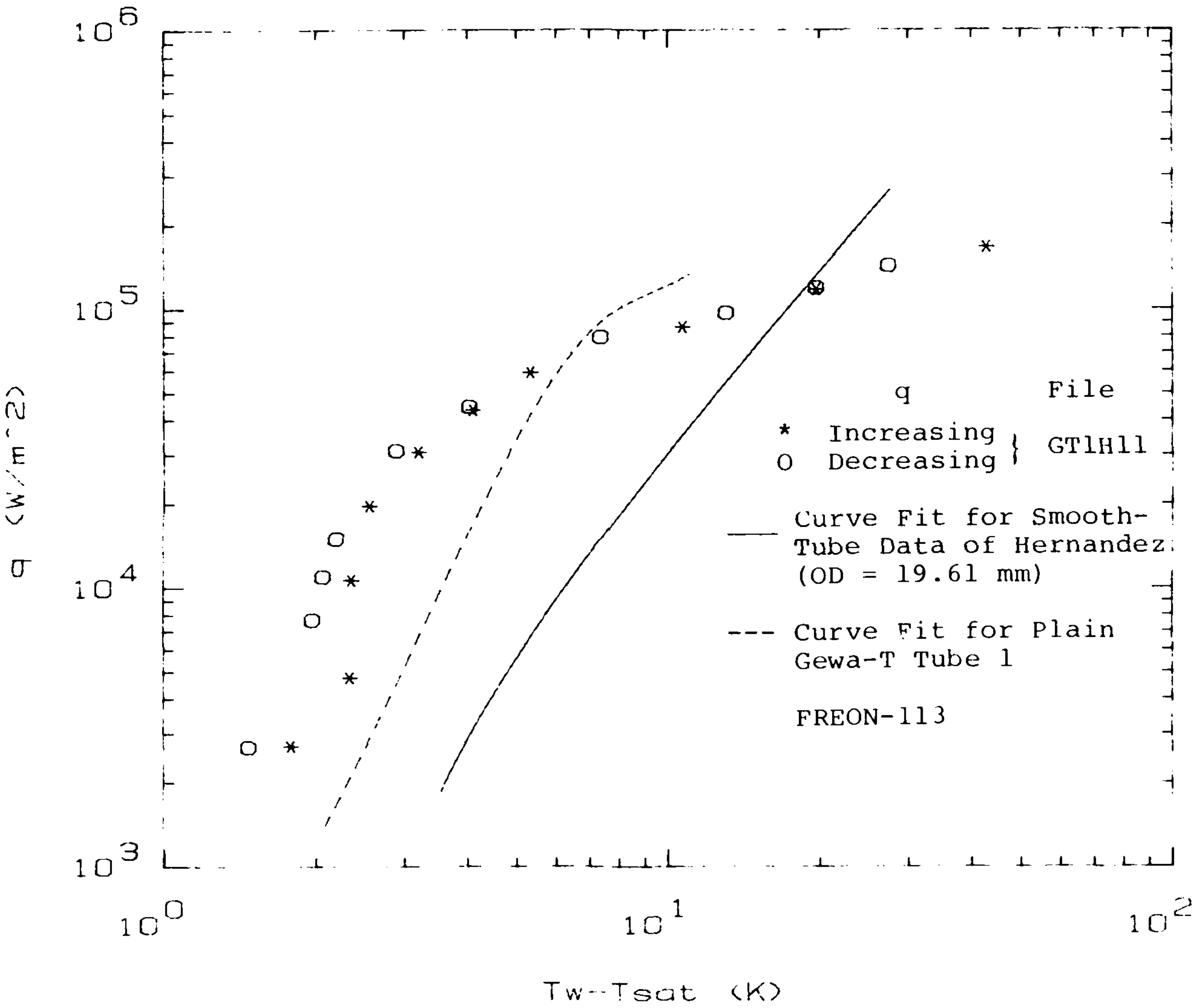


Figure 16. Effect of 60° × 8.5° Shroud on Plain Gewa-T Tube 1

TABLE 2

Summary of Results

DATA FILE NAME	TEST CONDITION	$q = 10{,}000\ W/m^2$ ΔT (°C)	h (W/m^2-K)	% INCR.	$q = 50{,}000\ W/m^2$ ΔT (°C)	h (W/m^2-K)	% INCR.	$q = 100{,}000\ W/m^2$ ΔT (°C)	h (W/m^2-K)	% INCR.
HERNANDEZ	SMOOTH	6.22	1607.7	-	12.7	3937.0	-	17.0	5882.4	-
GT1H05	PLAIN	3.68	2717.4	69.0	5.55	9009.0	128.8	8.77	11,402.5	93.8
GT1H09	PLAIN	3.65	2739.7	70.4	5.65	8849.6	124.8	8.61	11,614.4	97.4
GT1H11	Shrd. 60×8.5	1.76	5681.8	253.4	4.36	11,467.9	191.3	14.38	6954.1	18.20
GT1H12	Shrd. 60×30	1.94	5154.6	220.6	4.42	11,312.2	187.3	11.39	8779.6	49.30
GT1H13	Shrd. 60×60	2.50	4000.0	148.8	4.37	11,441.7	190.6	8.26	12,106.5	105.8
GT1H21	Shrd. 30×30	2.35	4255.3	164.7	4.54	11,013.2	179.7	15.07	6635.7	12.8
GT1H14	2 WIRES	2.25	4444.4	176.4	4.86	10,288.1	161.3	7.68	13,020.8	121.4
GT1H16	3 WIRES	1.41	7092.2	341.1	4.33	11,547.3	193.3	7.38	13,550.1	130.4
GT1H17	4 WIRES	1.58	6329.1	293.7	4.62	10,822.5	174.9	7.32	13,661.2	132.2
GT1H18	5 WIRES	1.62	6172.8	283.1	4.39	11,389.5	189.3	7.01	14,265.3	142.5
GT1H19	3 WR-Shr 60×8.5	1.07	9345.8	481.3	4.47	11,185.7	184.1	17.37	5757.1	-2.1
EXTRAPO-LATION	SMOOTH	6.57	1522.1	-	13.56	3687.3	-	18.55	5390.8	-
GT2H22	PLAIN	4.52	2212.4	45.4	6.94	7204.6	83.0	8.91	11,223.3	108.2
GT3H25	PLAIN	3.65	2739.8	80.0	6.12	8169.9	107.5	9.53	10,493.2	94.7
GT4H26	PLAIN	4.66	2145.9	41.0	8.34	5995.2	52.3	11.44	8441.3	62.2
GT5H27	PLAIN	5.13	1949.3	28.1	7.44	6720.4	70.7	9.38	10,661.0	97.8
GT5H28	PLAIN	5.15	1941.8	27.6	7.51	6605.0	67.8	9.56	10,460.3	94.0
GT5H29	PLAIN	3.24	3086.4	102.8	6.02	8305.7	111.0	8.5	11,764.7	118.2
GT7H30	PLAIN	5.18	1930.5	26.8	8.35	5988.0	52.1	10.61	9425.1	74.8

improvement. It can be seen that the values just presented are somewhat proportional to the summation of the two shroud angles. This behavior indicates that at very high heat fluxes, too small a shroud opening can obstruct the outflow of vapor and/or the inflow of liquid.

However, it must be noted that at all practical heat fluxes (10,000 to 80,000 W/m^2), all the shrouds resulted in a considerable enhancement in the boiling performance. The $60° \times 8.5°$ shroud appears to be the best selection for all practical heat fluxes.

I. PERFORMANCE OF GEWA-T TUBE WITH WIRE WRAPS IN INTER-FIN GAP

A series of runs were made with two to five 0.1 mm-diameter, copper wire wraps in the inter-fin gap of tube 1. Figures 17, 18, 19 and 20 show data with wire wraps of 2, 3, 4 and 5, respectively. It can be seen that all of these curves show increases in boiling performance at all wall superheats for decreasing heat flux values. This enhancement is most probably due to the increase in the nucleation sites within the inter-fin cavities. However, there is a small reduction in performance in the natural convection process probably owing to that wires could act as an insulating material due to the thermal contact resistance between the wires and the inter-fin area.

Table 2 also shows that the use of three wires resulted in the best performance increase. This combination resulted in a 341 percent improvement in the boiling heat-transfer

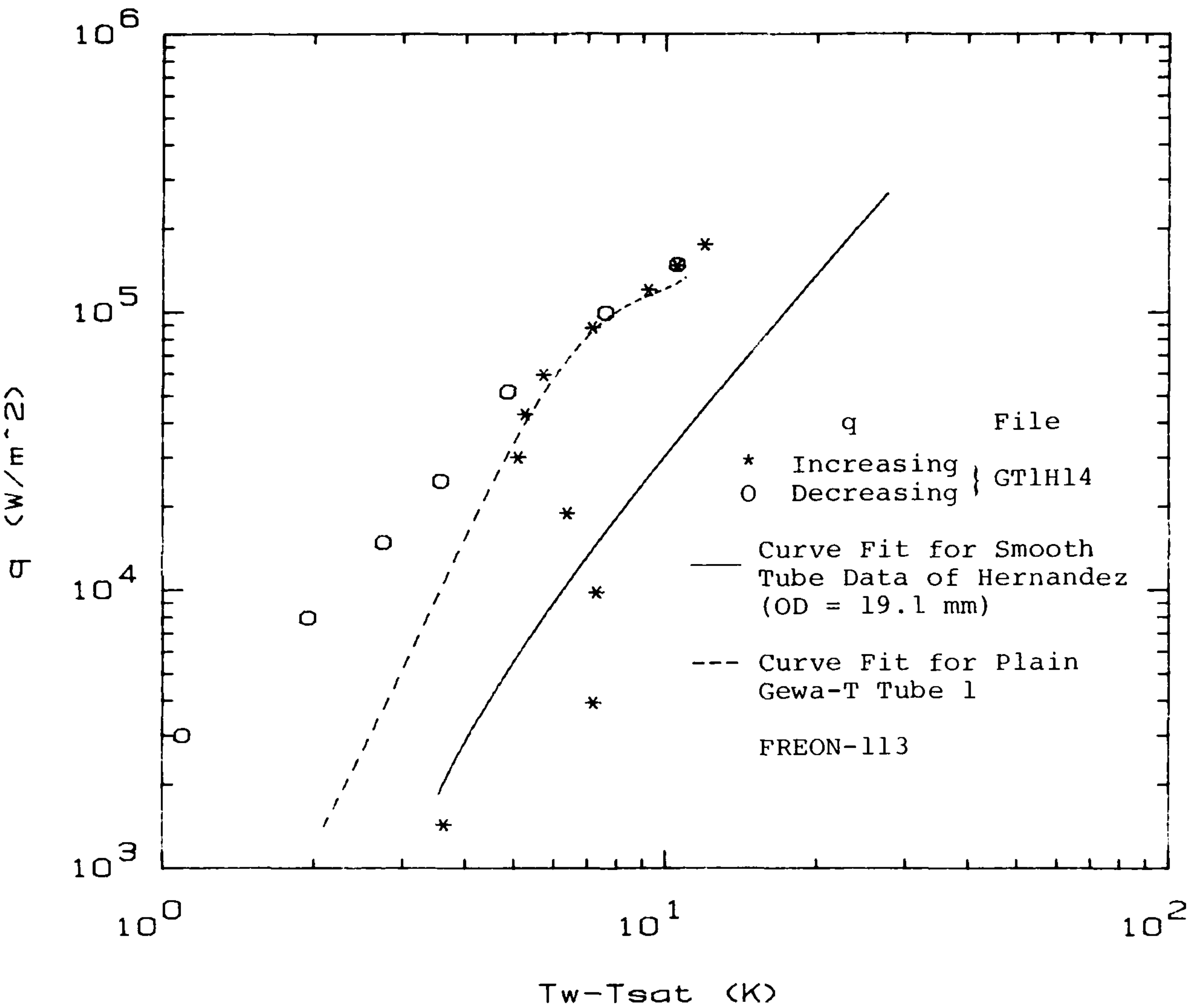


Figure 17. Effect of Two Wires Wrapped in the Inter-fin Cavity of Tube 1 on Boiling Performance

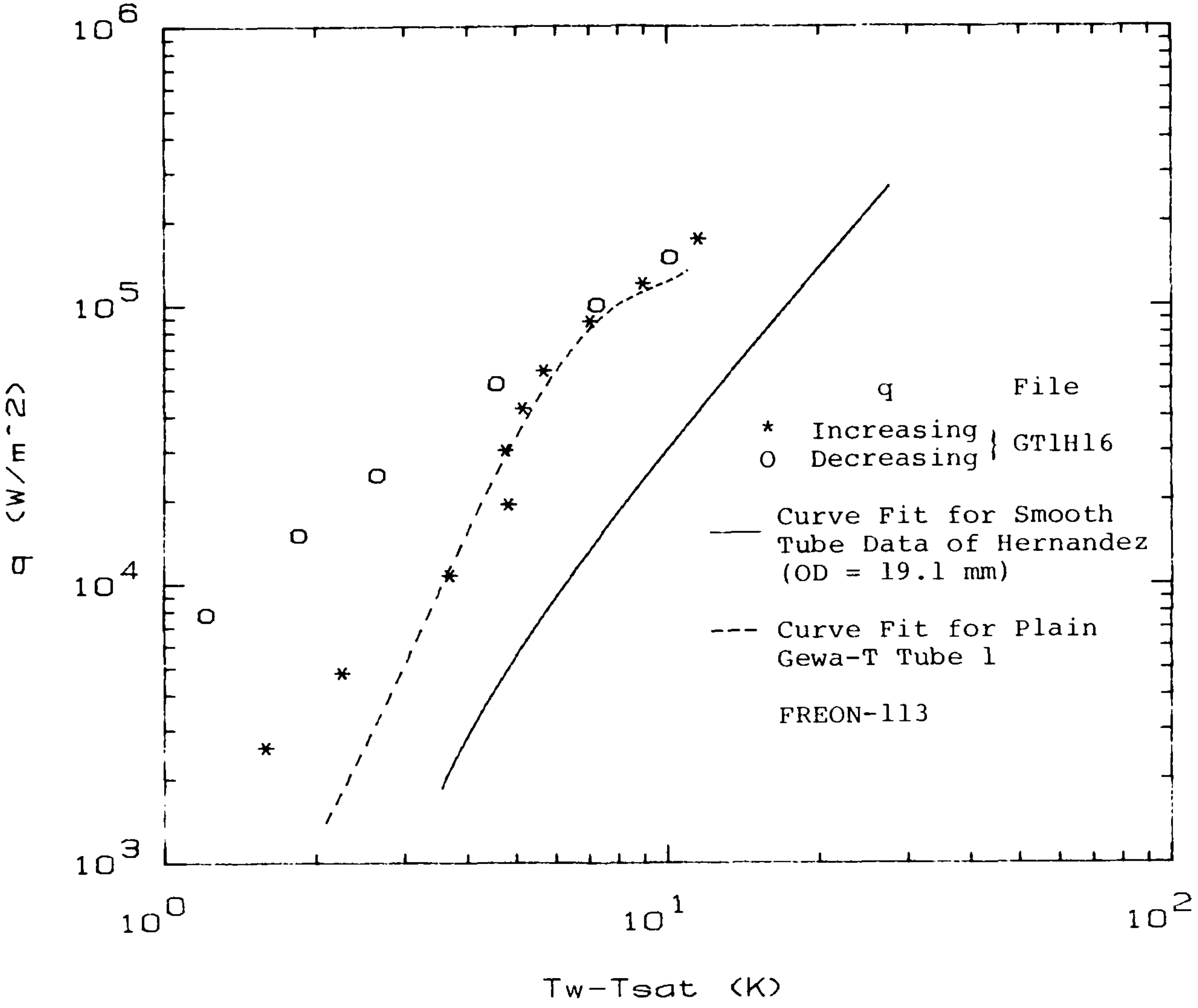


Figure 18. Effect of Three Wires Wrapped in the Inter-fin Cavity of Tube 1 on Boiling Performance

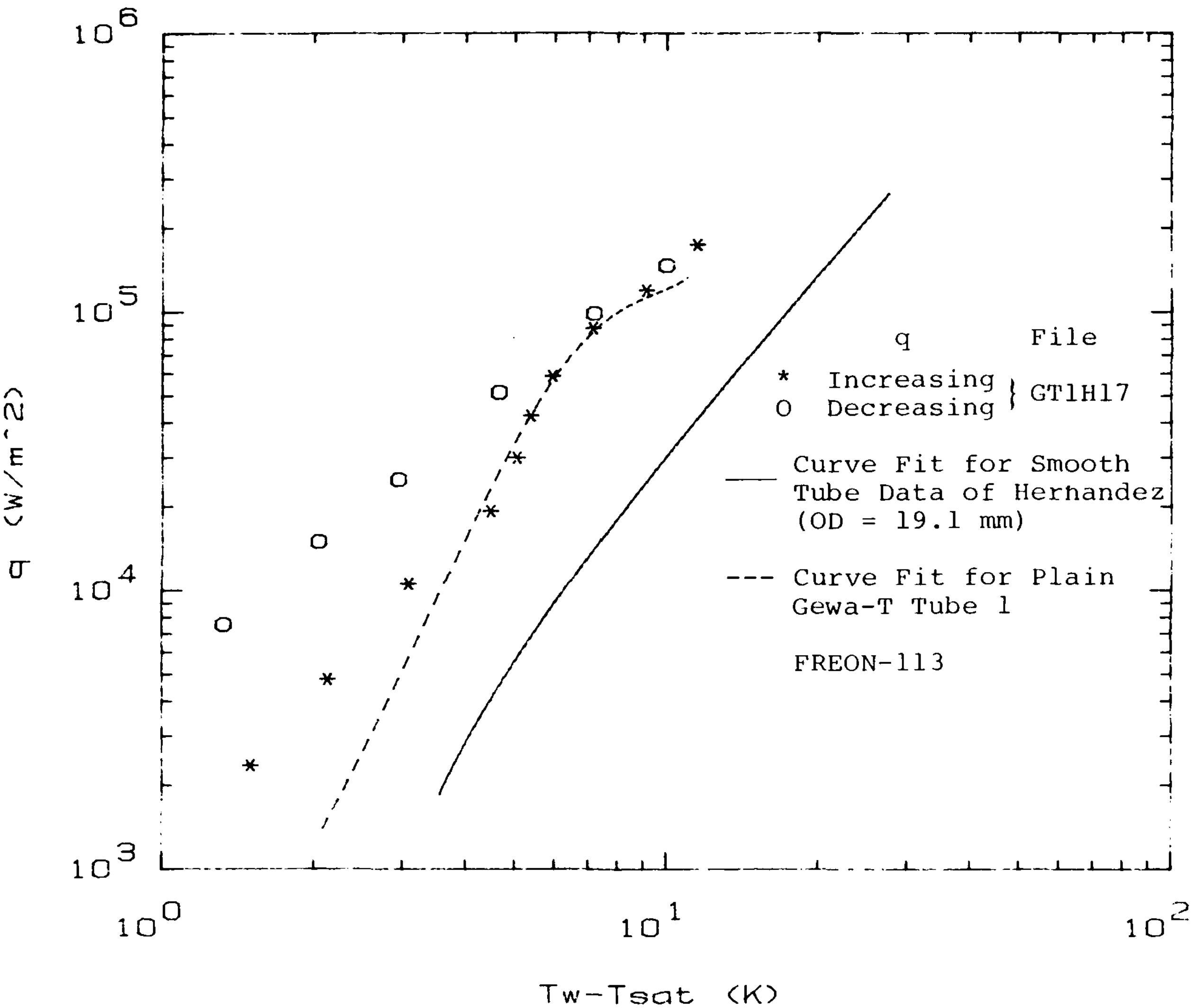


Figure 19. Effect of Four Wires Wrapped in the Inter-fin Cavity of Tube 1 on Boiling Performance

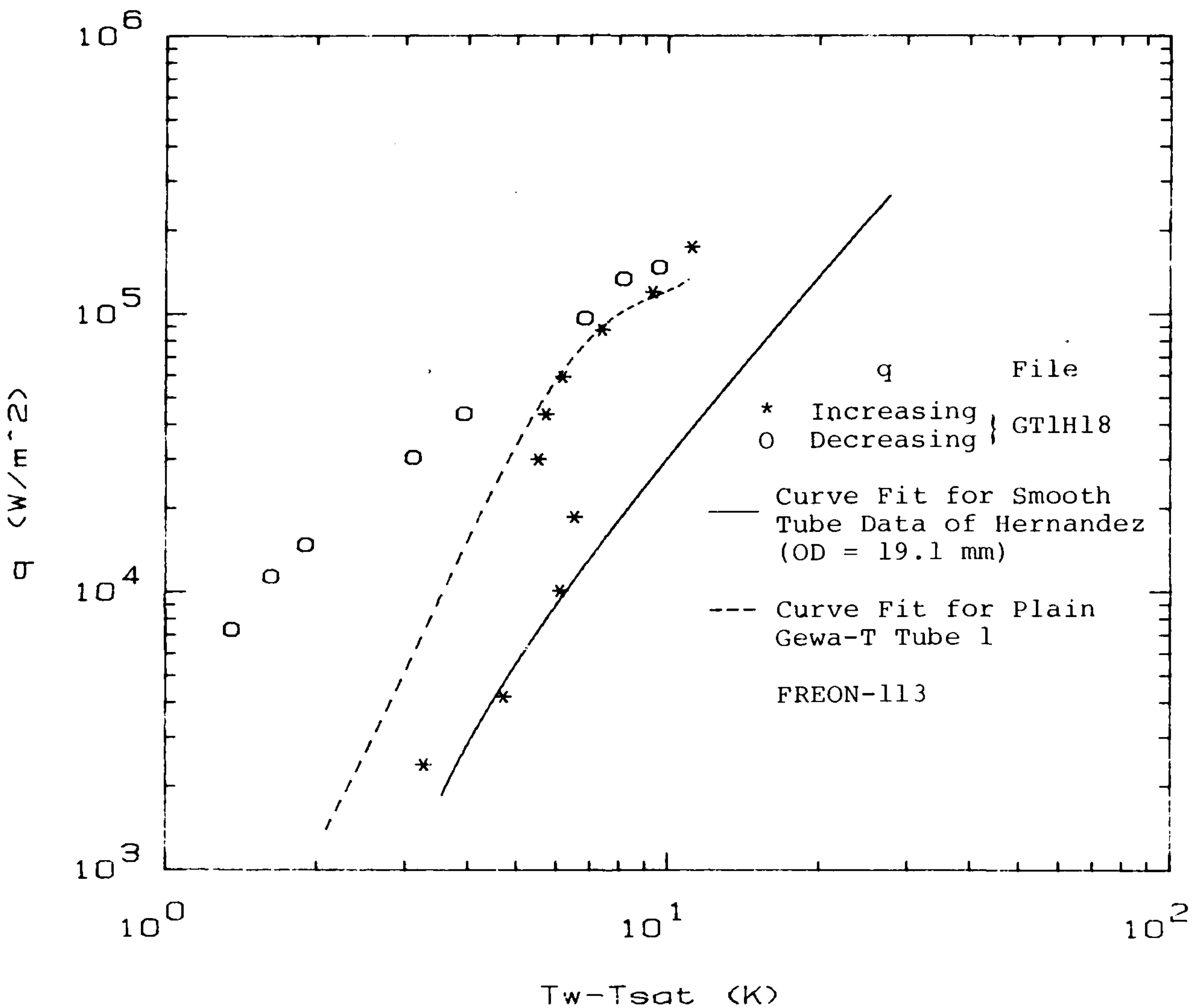


Figure 20. Effect of Five Wires Wrapped in the Inter-fin Cavity of Tube 1 on Boiling Performance

coefficient over the smooth tube value at a wall superheat of 1.4 K.

J. BOILING PERFORMANCE OF TUBE 1 WITH 60° × 8.5° SHROUD AND THREE WIRES

Data were also taken to combine the best shroud and the best number of wire wraps in the inter-fin channel. For this purpose, three wire wraps were added to the inter-fin channel and data were taken with the 60° × 8.5° shroud. As Figure 21 and Table 2 show, this combination resulted in a performance yet superior to the use of three wires or the use of 60° × 8.5° shroud separately. Table 2 shows that this arrangement resulted in 481 percent increase in the boiling heat-transfer coefficient over the smooth-tube value at a heat flux of 10,000 W/m^2.

K. EFFECTS OF FIN-TIP GAP AND FIN DENSITY OF GEWA-T TUBES ON BOILING PERFORMANCE

As described earlier, six Gewa-T surfaces were tested during this stage of the investigation; tubes 2, 3, and 4 had a fin density of 0.75 fins/mm, and tubes 5, 6 and 7 with a fin density of 1.02 fins/mm, and each set of tubes with fin-tip gaps of 0.15, 0.25 and 0.33 (see Table 1).

Micrographs were obtained for all enhanced surfaces on a Scanning Electron Microscope with a magnification of 50X. Figure 5 shows micrographs representing tubes 2 and 5. Computed surface areas and area ratios (with respect to smooth tube) are listed in Table 3.

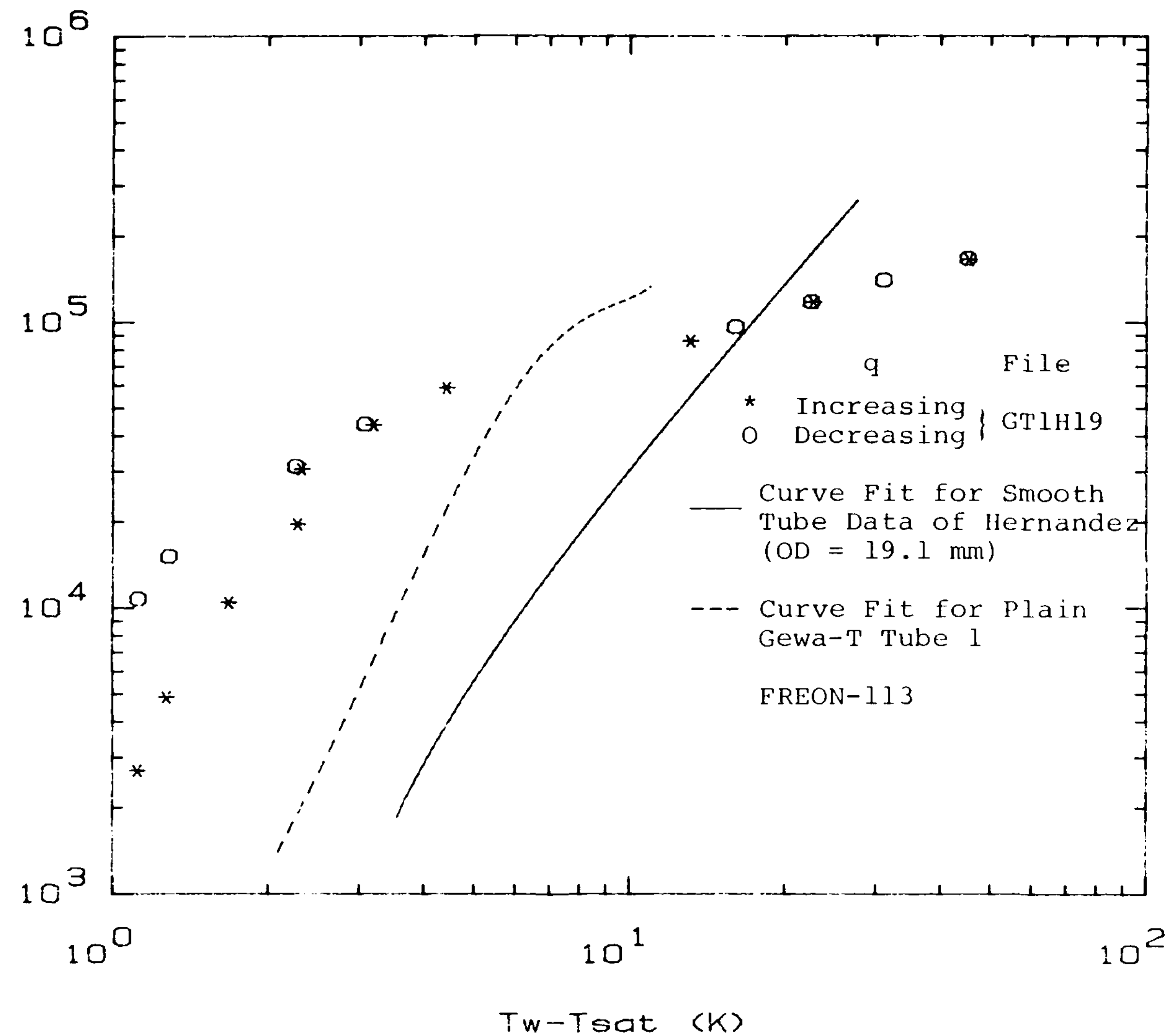


Figure 21. Effect of Three Wires Wrapped in the Inter-fin Cavity with 60° · 8.5° Shroud on Gewa-T Tube 1 on Boiling Performance

TABLE 3

Computed Surface Areas and Area Ratios

Tube	Cavity and Fin Surface Area (m^2)	Total Surface Area (m^2)	A_F/A_{SMOOTH}
1	240.47×10^{-6}	8.818×10^{-3}	---
2	288.01×10^{-6}	10.944×10^{-3}	2.74
3	305.34×10^{-6}	11.603×10^{-3}	2.91
4	285.07×10^{-6}	10.833×10^{-3}	2.72
5	256.04×10^{-6}	13.314×10^{-3}	3.60
6	248.61×10^{-6}	12.928×10^{-3}	3.50
7	267.91×10^{-6}	13.931×10^{-3}	3.77

It can be seen that tubes with a higher fin density (1.02 fins/mm) result in larger area ratios than the tubes with lower fin density (0.75 fins/mm). However, for a given fin density, the variation of the area ratio with fin-tip gap is not conclusive.

Figures 22 through 27 show data taken on tubes 2 through 7, respectively. These results are also listed in Table 2. This table shows that tube 6 (with a fin density of 1.02 fins/mm and a fin-tip gap of 0.25 mm) resulted in the highest heat flux at all values of wall superheat.

To investigate the effects of fin density and fin-tip gap on boiling heat-transfer performance, the results presented in Figures 22 through 27 (which showed variations of

APPENDIX A

DATA REDUCTION PROGRAM

```
1000! FILE NAME: DRP2
1010! DATE:        May 18, 1984
1020! REVISED:     July 20, 1984
1030!
1040  BEEP
1050  PRINTER IS 1
1060  PRINT USING "4X,""Select option:"""
1070  PRINT USING "6X,""0 Taking data or re-processing previous data"""
1080  PRINT USING "6X,""1 Plotting data"""
1090  INPUT Idp
1100  IF Idp=0 THEN CALL Main
1110  IF Idp=1 THEN CALL Plot
1120  END
1130  SUB Main
1140  COM /Cc/ C(7)
1150  DIM Emf(9),T(9)
1160  DATA 0.10086091,25727.94369,-767345.8295,78025595.81
1170  DATA -9247486589,6.97688E+11,-2.66192E+13,3.94078E+14
1180  READ C(*)
1190  BEEP
1200  INPUT "SELECT TUBE OD (0=0.75 in, 1=1.0 in)",Itd
1210  IF Itd=0 THEN
1220  D1=.0159        ! Diameter at thermocouple positions
1230  D2=.0193        ! Diameter of test section to the base of fins
1240  Di=.0183        ! Inside diameter of unenhanced ends
1250  Do=.0190        ! Outside diameter of unenhanced ends
1260  L=.0490         ! Length of enhanced surface
1270  Lu=.0319        ! Length of unenhanced surface at the ends
1280  ELSE
1290  D1=.01796
1300  D2=.0231
1310  Di=.02032
1320  Do=.02222
1330  L=.0508
1340  Lu=.03175
1350  END IF
1360  Lc=Lu+(Do-Di)/2
1370  A=PI*(Do^2-Di^2)/4
1380  P=PI*Do
1390  Kcu=385         ! Thermal conductivity of Copper
1400  PRINTER IS 701
1410  CLEAR 709
1420  BEEP
1430  INPUT "ENTER MONTH, DATE AND TIME (MM:DD:HH:MM:SS)",Date$
1440  OUTPUT 709;"TD";Date$
1450  OUTPUT 709;"TD"
1460  ENTER 709;Date$
1470  PRINT "                  Month, date and time :";Date$
1480  PRINT
1490  PRINT USING "10X,""NOTE: Program name : BOIL"""
1500  BEEP
1510  INPUT "ENTER DISK NUMBER",Dn
1520  PRINT USING "16X,""Disk number  = "",ZZ";Dn
1530  BEEP
1540  INPUT "ENTER INPUT MODE (0=3054A,1=FILE)",Im
1550  Im=Im+1
1560  IF Im=1 THEN
1570  BEEP
1580  INPUT "GIVE A NAME FOR THE RAW DATA FILE",D_file$
1590  PRINT USING "16X,""New file name: "",14A";D_file$
```

VI. RECOMMENDATIONS

(1) Investigate the reason for the existence of a longitudinal temperature profile and take appropriate steps to alleviate this problem.

(2) Study the possibility of adding roughness in the inter-fin cavity of Gewa-T tubes. This operation should be performed before the rolling process to obtain the T-shaped fins.

(3) Repeat runs made during this investigation with water as the boiling medium.

This combination resulted in a 481 percent increase in boiling heat-transfer coefficient at a wall superheat of 1.1 K.

(6) The use of wires appears to be superior to the use of more expensive shrouds.

(7) Among the fin-tip gaps of 0.15, 0.25 and 0.35 mm tested, the 0.25 mm gap resulted in the best boiling performance.

(8) The Gewa-T tubes with a fin density of 1.02 fins/mm performed superior to a fin density of 0.75 fins/mm. At a wall superheat of 3.5 K, the lower-fin-density tube increased the boiling heat-transfer coefficient by 80 percent, while the higher-fin-density tube increased by 103 percent.

V. CONCLUSIONS

(1) Experimental observations revealed a non-uniform temperature distribution along test section. The bubble generation was seen to start at the two ends and migrate into the center as the heat flux was increased. No clear explanation is possible for this temperature distribution.

(2) Results of the present investigation showed very good repeatability with the results of Hernandez obtained under similar conditions.

(3) The use of shrouds improved the boiling heat-transfer coefficient of Gewa-T surface at all practical heat fluxes. However, a considerable reduction in heat-transfer coefficient was seen at very high heat fluxes ($> 100{,}000$ W/m^2). Among the four shrouds tested, the shroud with $60° \times 8.5°$ openings resulted in the best performance verifying Hernandez's results. This shroud resulted in a 253 percent increase in boiling heat-transfer coefficient, over smooth tube, at a wall superheat of 1.8 K.

(4) The use of 0.1 mm-diameter wires (2 to 5) improved boiling heat-transfer coefficient at all practical heat fluxes. The use of 3 wires resulted in the best performance with a 341 percent increase in the heat-transfer coefficient (over the smooth-tube value) at a wall superheat of 1.4 K.

(5) Further improvement in boiling performance was obtained by using the $60° \times 8.5°$ shroud with three wire wraps.

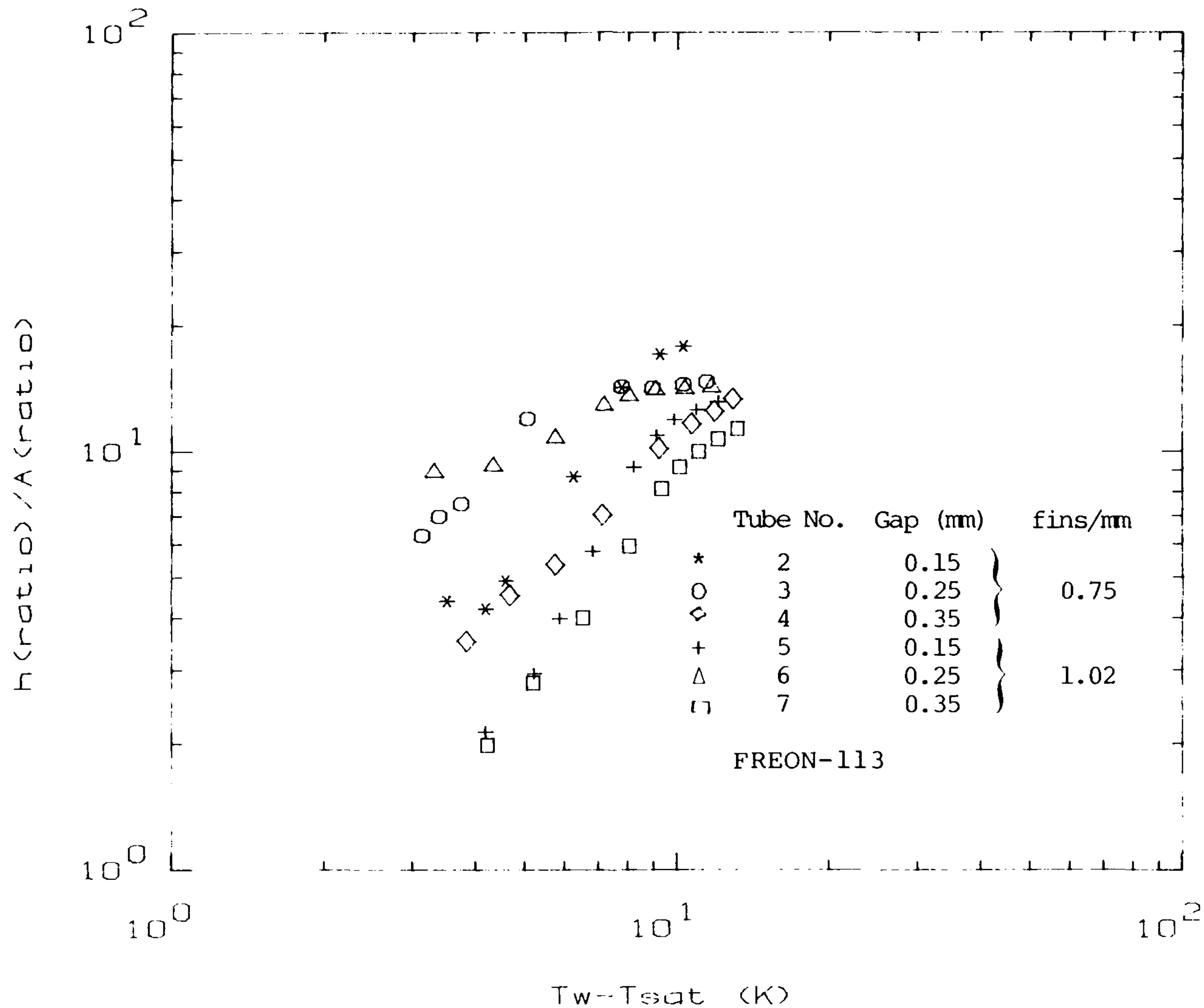


Figure 29. Enhancement Ratios of Gewa-T Tubes over Smooth Tube Based on Actual Area

explanations given above should be valid for the poorer performance of tubes 2 and 4.

It can be seen that all tubes show smaller enhancement ratios at low wall superheats, and the above-mentioned differences in performance are more pronounced. On the other hand, all the tubes show greater enhancements at larger wall superheats and they seem to approach a common enhancement ratio. The increase in enhancement is believed to be due to the increased chimney effect, which helps to remove vapor bubbles from channels resulting in a thinner liquid film at the hot boiling surface. When heat flux is further increased, vapor can provide a blanket around the hot surface, which may eventually result in a critical heat flux. It appears that the critical heat flux is not strongly dependent on the fin density or fin-tip gap.

The results shown in Figure 28 were also replotted as shown in Figure 29 to further study the performance of these six tubes. This figure shows the variation in enhancement ratio obtained beyond the area increase (i.e., the ratio of heat-transfer enhancement/area ratio) as a function of wall superheat. It is evident that obviously all tubes resulted in enhancement ratios beyond the area increase.

In summary, a fin density of 1.02 fins/mm resulted in performance superior to a fin density of 0.75 fins/mm. Also, in both cases, a fin-tip gap of 0.25 mm resulted in the best performance.

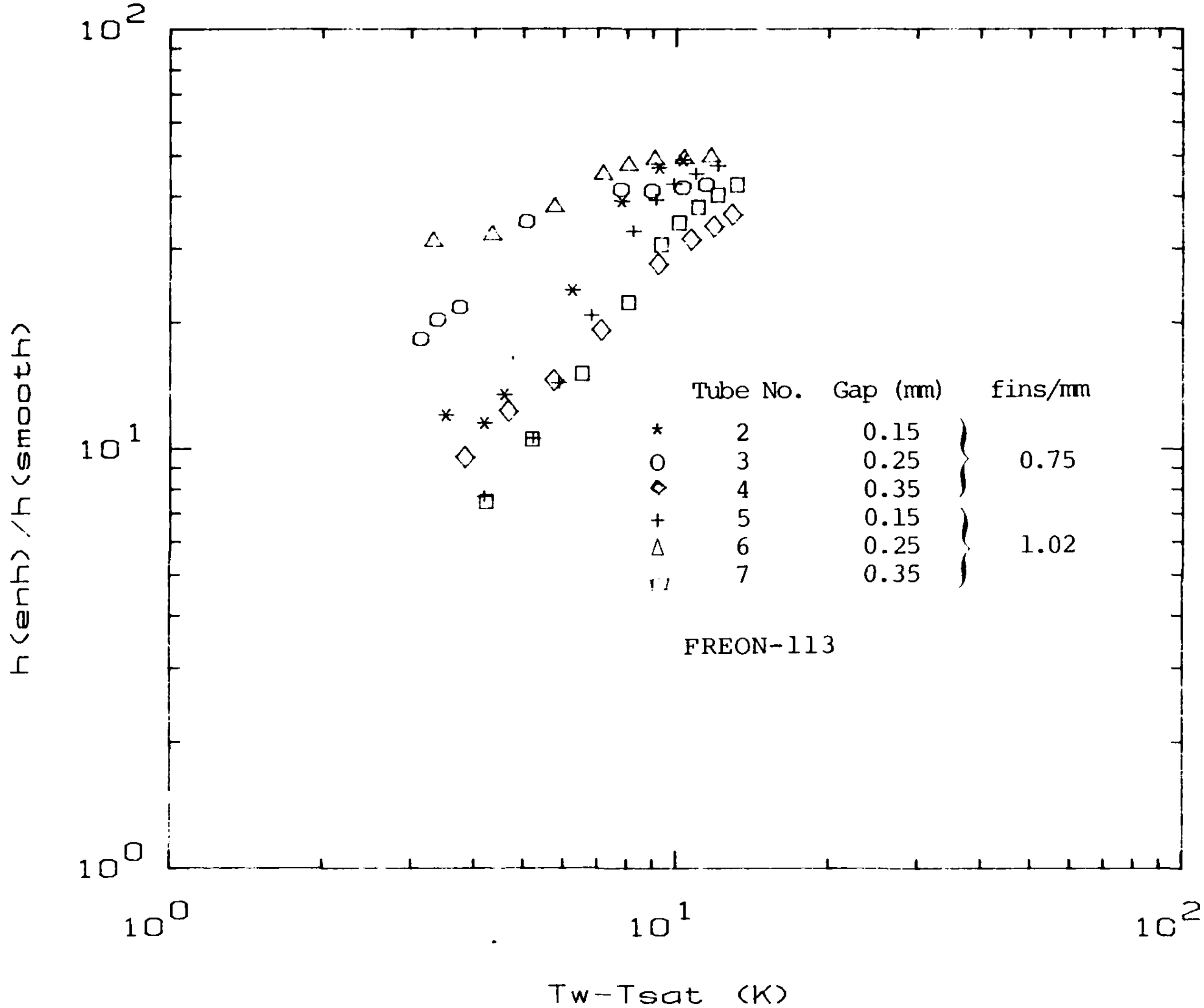


Figure 28. Enhancement Ratios of Gewa-T Tubes over Smooth Tubes Based on Nominal Area

heat flux as a function of wall superheat) were re-plotted on a different basis as shown in Figure 28. This figure shows the variations of the enhancement of heat-transfer coefficient, over the smooth-tube value, with wall superheat for all six tubes. It is evident that tube 6, which has a fin density of 1.02 fins/mm and a fin-tip gap of 0.25 mm results in the maximum enhancement at all wall superheat values.

Tube 5 shows poorer performance than tube 6. This poorer performance is believed to be due to the fact that the smaller fin-tip gap disrupts the flow of liquid into the channels and/or the vapor flow out of the channels. Tube 7 (which has the largest fin-tip gap of 0.35 mm) shows the poorest performance among the set of these tubes with a fin density of 1.02 fins/mm. As visually observed during experimental runs, this tube ejected a considerable portion of vapor bubbles radially along tube periphery. This premature departure of vapor bubbles results in a lower performance by producing a thicker liquid film at the hot surface.

In comparison with the results for tubes 5, 6 and 7, the curves representing tubes 2, 3 and 4 show that for a given fin-tip gap, the performance has decreased with decrease in fin density. This observation is believed to be due to the presence of a greater number of channels (for a given length of tube) in the higher-fin-density tubes. Again, for a fin density of 0.75 fins/mm, the 0.25 mm fin-tip gap (tube 3) resulted in the best boiling performance. The same

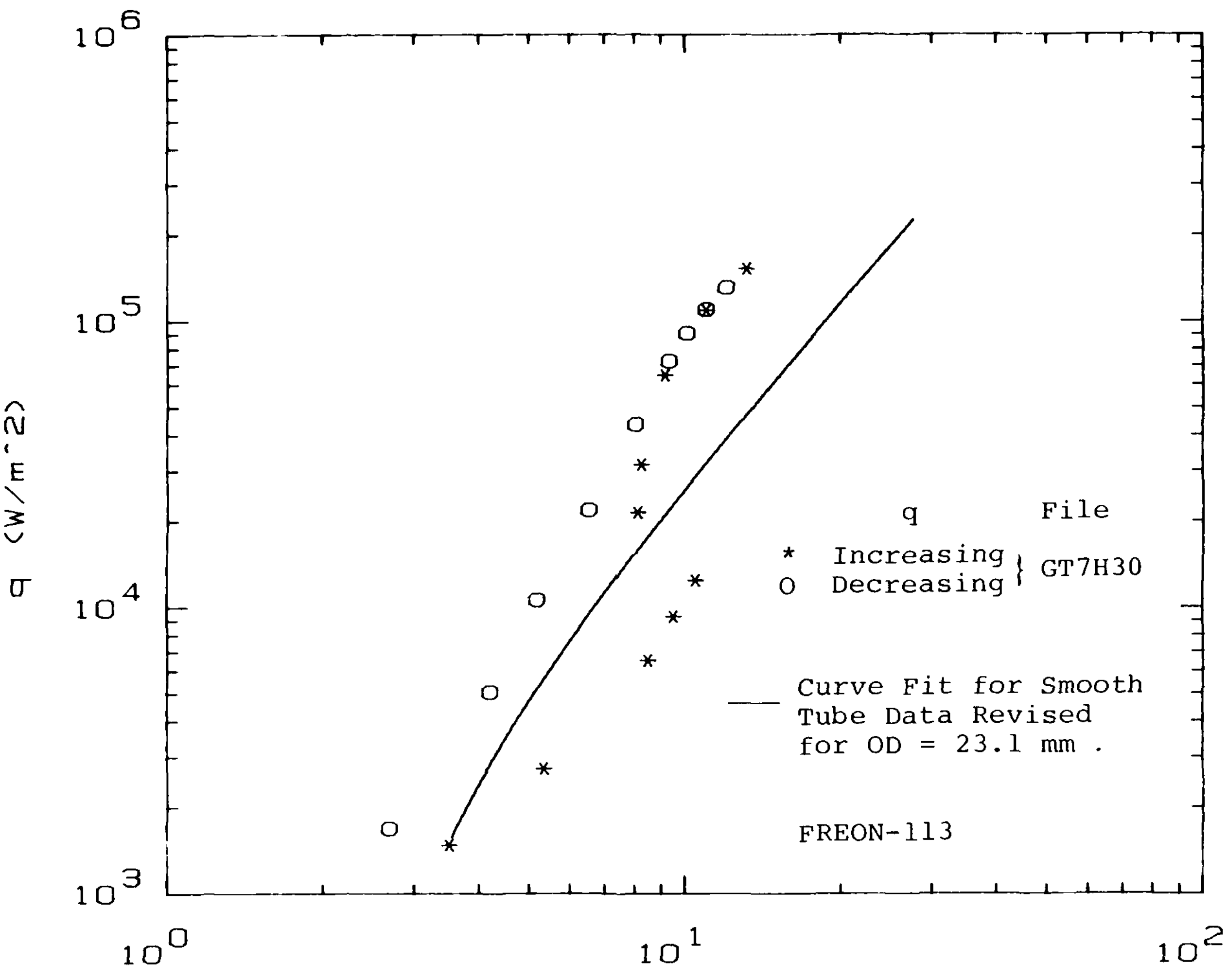


Figure 27. Boiling Performance of Gewa-T Tube 7

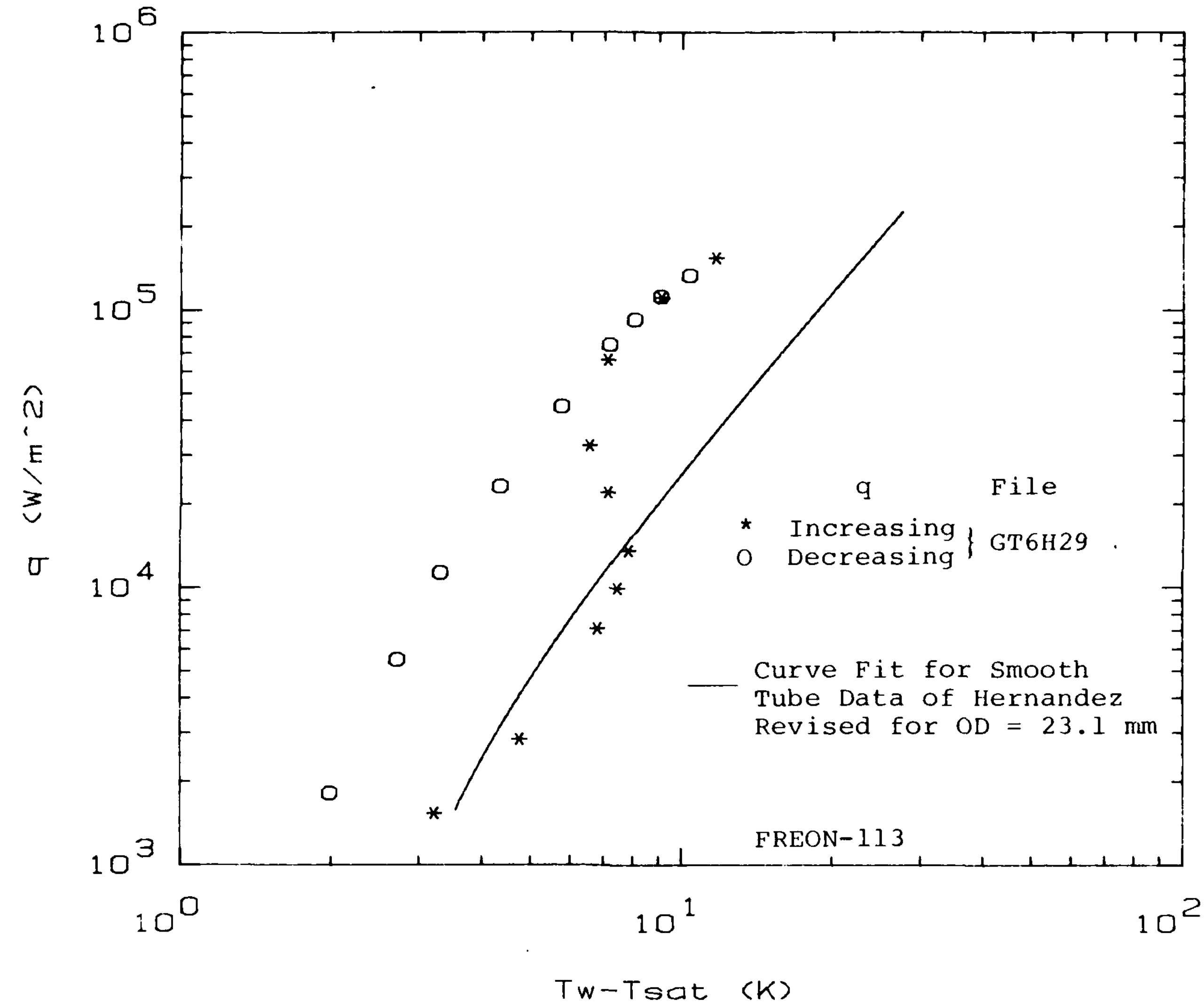


Figure 26. Boiling Performance of Gewa-T Tube 6

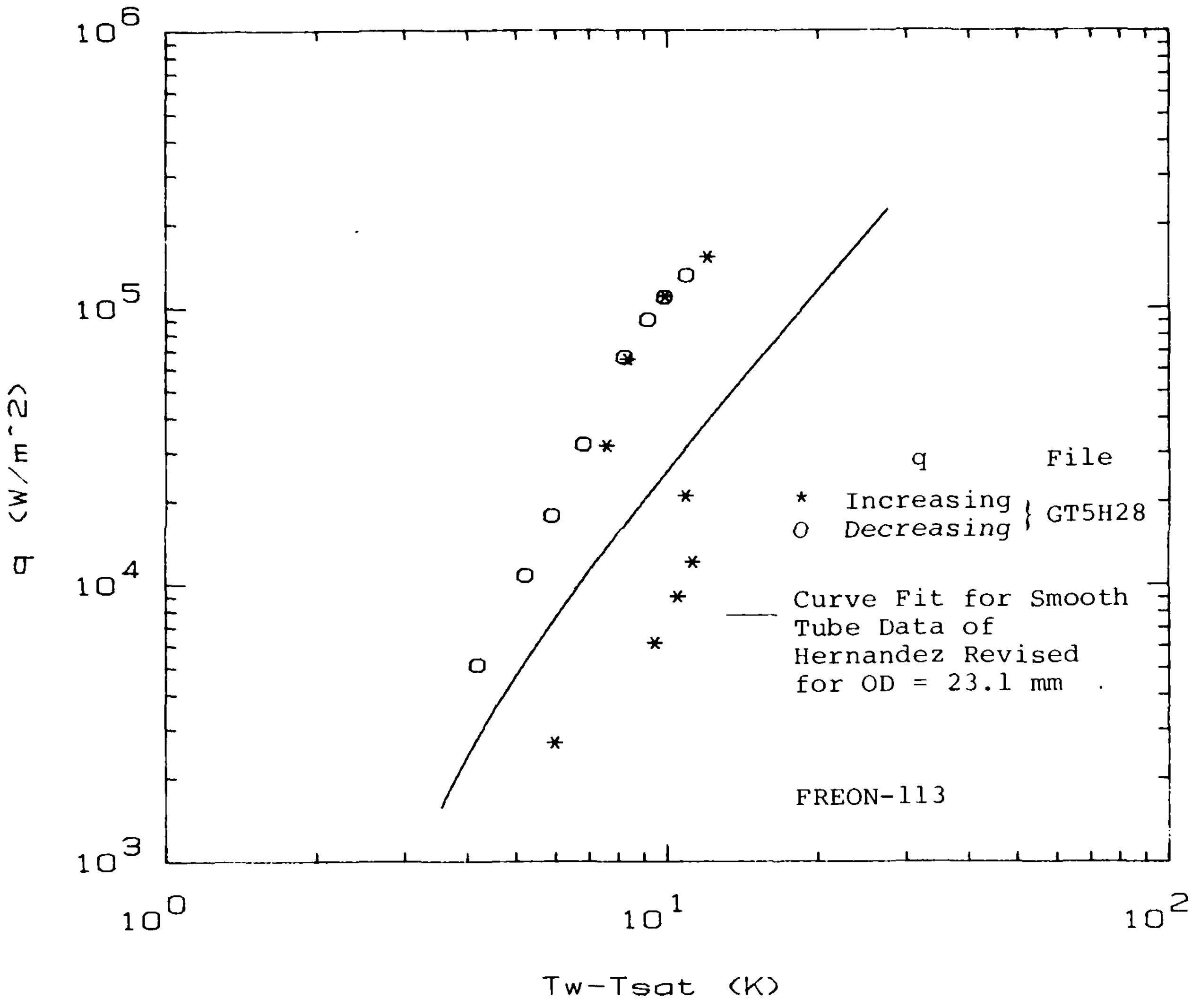


Figure 25. Boiling Performance of Gewa-T Tube 5

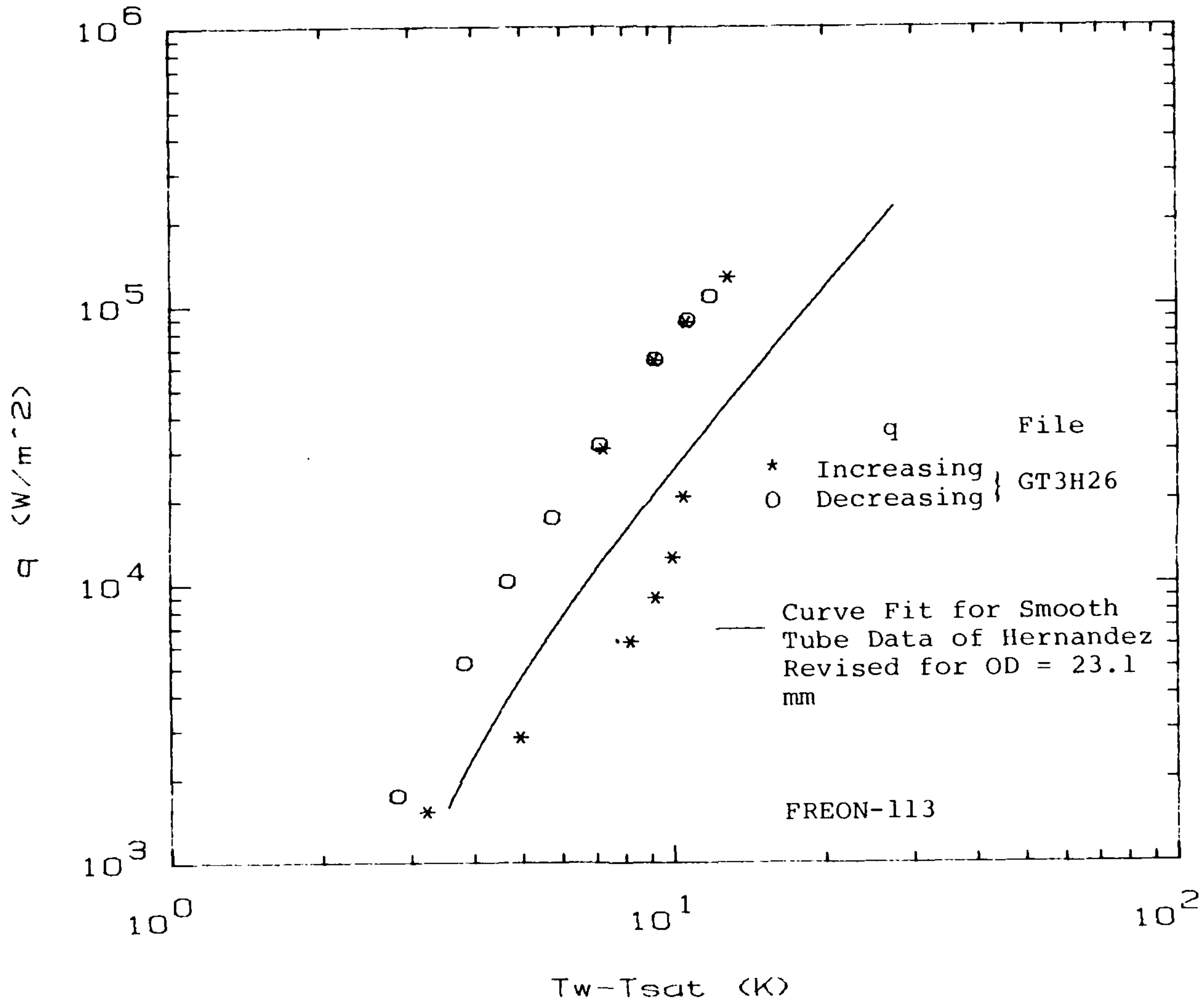


Figure 24. Boiling Performance of Gewa-T Tube 4

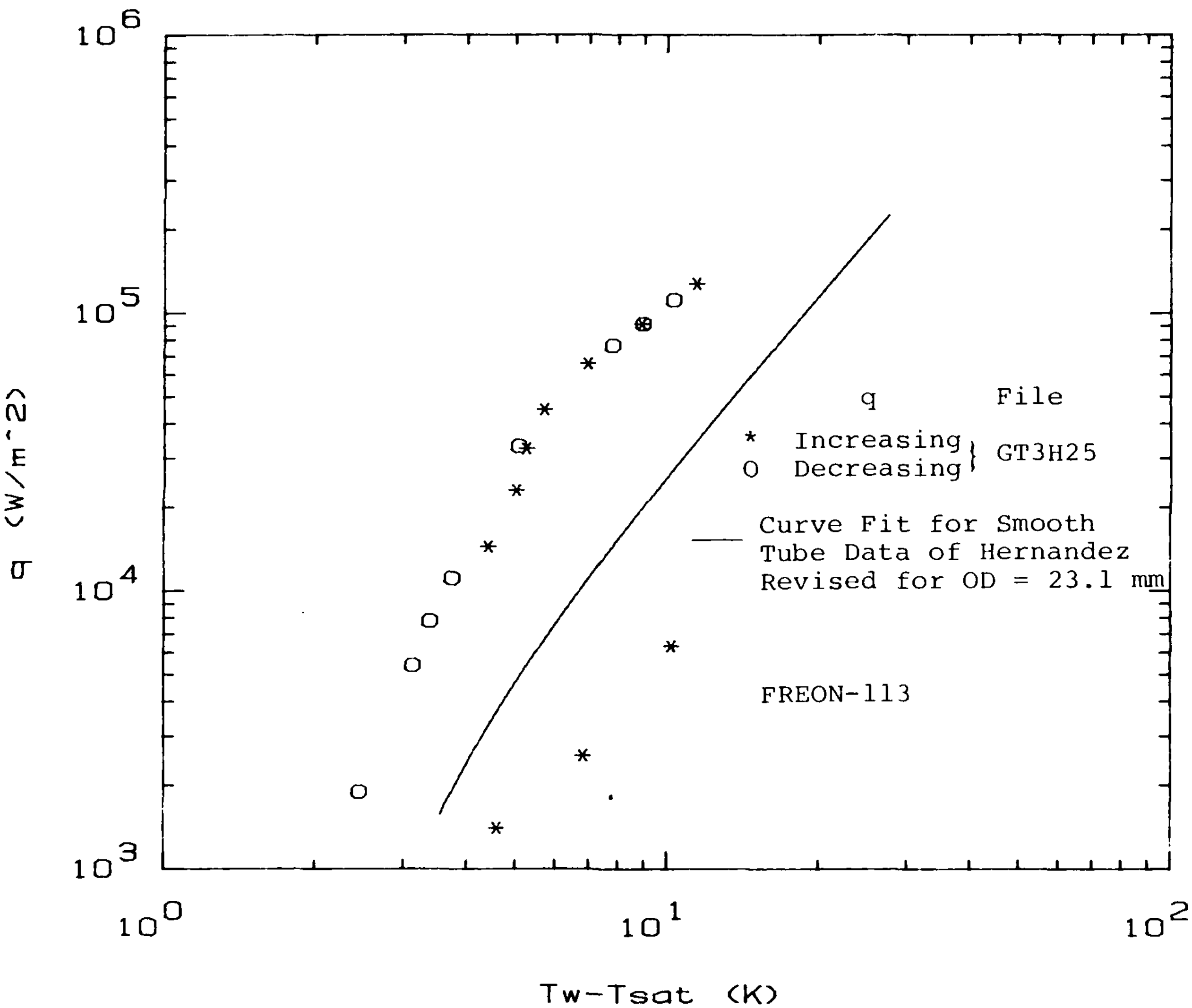


Figure 23. Boiling Performance of Gewa-T Tube 3

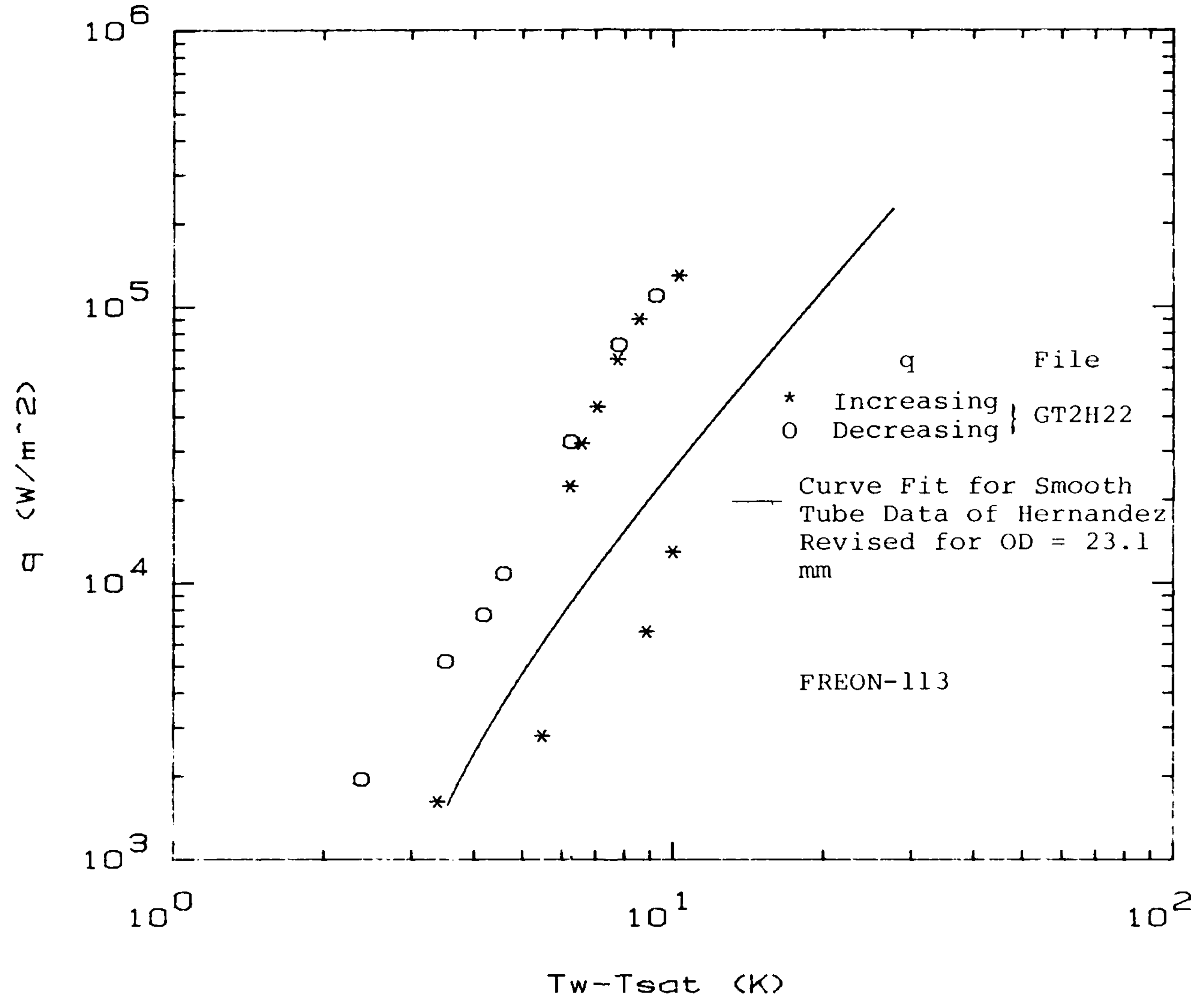


Figure 22. Boiling Performance of Gewa-T Tube 2

```
1600  BEEP
1610  INPUT "ENTER FLUID (0=FREON-113,1=WATER)",Cf
1620  Cf=Cf+1
1630  CREATE BDAT D_file$,20
1640  ASSIGN @File TO D_file$
1650  OUTPUT @File:Date$,Cf
1660  ELSE
1670  BEEP
1680  INPUT "GIVE THE NAME OF THE EXISTING DATA FILE",D_file$
1690  BEEP
1700  Ihc=0
1710  INPUT "ENTER 1 IF USING FILE NUMBER < 25",Ihc
1720  PRINT USING "16X,""Old file name: "",14A":D_file$
1730  BEEP
1740  INPUT "ENTER THE NUMBER OF RUNS STORED",Nrun
1750  ASSIGN @File TO D_file$
1760  ENTER @File:Dold$,Cf
1770  PRINT USING "16X,""This output for file created at "",14A":Dold$
1780  END IF
1790  BEEP
1800  INPUT "GIVE A NAME FOR PLOT FILE",P_file$
1810  CREATE BDAT P_file$,5
1820  ASSIGN @Plot TO P_file$
1830  IF Cf=1 THEN PRINT USING "16X,""Fluid         : Freon-113"""""
1840  IF Cf=2 THEN PRINT USING "16X,""Fluid         : Water"""""
1850  BEEP
1860  INPUT "ENTER OUTPUT VERSION (1=SHORT,0=LONG)",Iov
1870  IF Iov=0 THEN Iov=2
1880  BEEP
1890  INPUT "ENTER NUMBER OF DEFFECTIVE TCS (0=DEFAULT)",Idtc
1900  IF Idtc>0 THEN
1910  BEEP
1920  IF Idtc=1 THEN
1930  INPUT "ENTER DEFFECTIVE TC LOCATION",Ldtc1
1940  PRINT USING "16X,""TC is deffective at location "",D":Ldtc1
1950  END IF
1960  IF Idtc=2 THEN
1970  INPUT "ENTER DEFFECTIVE TC LOCATIONS",Ldtc1,Ldtc2
1980  PRINT USING "16X,""TC are deffective at locations "",D,4X,D":Ldtc1,Ldtc2
1990  END IF
2000  END IF
2010  IF Im=1 THEN
2020  BEEP
2030  INPUT "ENTER BAROMETRIC PRESSURE (inHg)",Patm
2040  Patm=Patm+2*1500/13500
2050  OUTPUT @File:Patm
2060  ELSE
2070  ENTER @File:Patm
2080  IF Ihc=1 THEN Patm=Patm+2*1500/13500
2090  END IF
2100  Patm=Patm*25.4*14.696/760
2110  Tsat=FNTsat(Cf,Patm)
2120  PRINT USING "16X,""Saturation temperature = "",3D.DD,"" (Deg C)""":Tsat
2130  J=1
2140  Sx=0
2150  Sy=0
2160  Sxs=0
2170  Sxy=0
2180  BEEP
2190  INPUT "HIT CONTINUE WHEN READY",Ok
```

```
2200 Repeat: !
2210  IF Im=' THEN
2220  OUTPUT 709:"AR AF29 AL38 VR5"
2230  FOR I=0 TO 9
2240  OUTPUT 709:"AS SA"
2250  Sum=0
2260  FOR Ji=1 TO 20
2270  ENTER 709:E
2280  Sum=Sum+E
2290  NEXT Ji
2300  Emf(I)=Sum/20
2310  NEXT I
2320  BEEP
2330  INPUT "ENTER HEATER VOLTAGE, Vh (V)",Vh
2340  BEEP
2350  INPUT "ENTER RESISTOR VOLTAGE, Vr (V)",Vr
2360  ELSE
2370  ENTER @File:Emf(*),Vr,Vh
2380  END IF
2390  Twa=0
2400  FOR I=0 TO 9
2410  IF Idtc>0 THEN
2420  IF (I+1)=Ldtc1 OR (I+1)=Ldtc2 THEN
2430  T(I)=99.99
2440  GOTO 2490
2450  END IF
2460  END IF
2470  T(I)=FNTvsv(Emf(I))
2480  IF I<8 THEN Twa=Twa+T(I)
2490  NEXT I
2500  Twa=Twa/(8-Idtc)
2510  Tld=T(9)
2520  Tv=T(8)
2530  Rho=FNRho(Cf,Tsat)
2540  Mu=FNMu(Cf,Tsat)
2550  K=FNK(Cf,Tsat)
2560  Cp=FNCp(Cf,Tsat)
2570  Beta=FNBeta(Cf,Tsat)
2580  Ni=Mu/Rho
2590  Alpha=K/(Rho*Cp)
2600  Pr=Ni/Alpha
2610  Amp=Vr/2.031
2620  Q=Vh*Amp
2630  Tw=Twa-Q*LOG(D2/D1)/(2*PI*Kcu*L)
2640  Thetab=Tw-Tsat
2650  IF Thetab<0 THEN
2660  BEEP
2670  DISP "TWALL<TSAT (HIT CONTINUE WHEN READY)"
2680  PAUSE
2690  GOTO 2210
2700  END IF
2710  Hbar=190
2720  Fe=(Hbar*P/(Kcu*A)) .5*Lc
2730  Tanh=FNTanh(Fe)
2740  Theta=Thetab*Tanh/Fe
2750  Xx=(9.81*Beta*Thetab*Do 3*Tanh/(Fe*Ni*Alpha)) .166667
2760  Yy=(1+(.559/Pr) (9/16)) (8/27)
2770  Hbarc=K/Do*(.6+.387*Xx/Yy) 2
2780  IF ABS((Hbar-Hbarc)/Hbarc)>.001 THEN
2790  Hbar=(Hbar+Hbarc)*.5
2800  GOTO 2720
```

```
2810  END IF
2820  Q1=(Hbar*P*Kcu*A)^.5*Thetab*Tann
2830  Qc=Q-2*Q1
2840  As=PI*D2*L
2850  Qdp=Qc/As
2860  IF Iov=1 THEN PRINT USING "10X.DD.2X.3D.DD.2X.2(Z.4DE.2X).DD.DD":J.Twa.Hba
r.Qdp.Thetab
2870  IF Iov=2 THEN
2880  PRINT USING "10X.""Data set number = "".DD":J
2890  PRINT USING "10X.""Position number :   1      2      3      4      5      6
 7      8"""
2900  PRINT USING "10X.""Temperature      : "".8(DD.DD.1X)":T(0).T(1).T(2).T(3).T
(4).T(5).T(6).T(7)
2910  PRINT USING "10X.""Tliqd    Tvapr     Twa       Hbar          Qdp       Thetab"""
2920  PRINT USING "9X.3(3D.DD.2X).2(Z.4DE.2X).2D.DD":Tld.Tv.Twa.Hbar.Qdp.Thetab
2930  END IF
2940  IF Im=1 THEN
2950  BEEP
2960  INPUT "OK TO STORE THIS DATA SET (1=Y.0=N)?".Ok
2970  END IF
2980  IF Ok=1 OR Im=2 THEN J=J+1
2990  IF Ok=1 AND Im=1 THEN OUTPUT @File:Emf(*).Vr.Vh
3000  IF Im=2 OR Ok=1 THEN OUTPUT @Plot:Qdp.Thetab
3010  IF Im=1 THEN
3020  BEEP
3030  INPUT "WILL THERE BE ANOTHER RUN (1=Y.0=N)?".Go_on
3040  Nrun=J
3050  IF Go_on<>1 THEN 3110
3060  WAIT 300
3070  IF Go_on=1 THEN Repeat
3080  ELSE
3090  IF J<Nrun+1 THEN Repeat
3100  END IF
3110  IF Im=1 THEN
3120  BEEP
3130  PRINT
3140  PRINT USING "10X.""NOTE: "".ZZ."" data runs were stored in file "".10A":J-
1.D_file$
3150  END IF
3160  BEEP
3170  PRINT
3180  PRINT USING "10X.""NOTE: "".ZZ."" X-Y pairs were stored in plot data file
"".10A":J-1.P_file$
3190  ASSIGN @File TO *
3200  ASSIGN @Plot TO *
3210  BEEP
3220  INPUT "LIKE TO PLOT DATA (1=Y.0=N)?".Ok
3230  IF Ok=1 THEN
3240  CALL Plot
3250  END IF
3260  SUBEND
3270  DEF FNMu(Cf.T)
3280  IF Cf=1 THEN
3290! RANGE: 28 TO 50 DEG C
3300  Mu=8.9629819E-4-T*(1.1094609E-5-T*5.566829E-8)
3310  END IF
3320  IF Cf=2 THEN
3330  A=247.8/(T+133.15)
3340  Mu=2.4E-5*10^A
3350  END IF
```

```
3360  RETURN Mu
3370  FNEND
3380  DEF FNCp(Cf,T)
3390  IF Cf=1 THEN
3400! RANGE: 0 TO 80 DEG C
3410  Cp=9.2507273E-1+T*(9.3400433E-4+1.7207792E-5*T)
3420  END IF
3430  IF Cf=2 THEN
3440  Cp=4.21120858-T*(2.26826E-3-T*(4.42361E-5+2.71428E-7*T))
3450  END IF
3460  RETURN Cp*1000
3470  FNEND
3480  DEF FNRho(Cf,T)
3490  IF Cf=1 THEN
3500! RANGE: 30 TO 80 DEG C
3510  Ro=1.6207479E+3-T*(2.2186346+T*2.3578291E-3)
3520  END IF
3530  IF Cf=2 THEN
3540  Ro=999.52946+T*(.01269-T*(5.482513E-3-T*1.234147E-5))
3550  END IF
3560  RETURN Ro
3570  FNEND
3580  DEF FNPr(Cf,T)
3590  Pr=FNCp(Cf,T)*FNMu(Cf,T)/FNK(Cf,T)
3600  RETURN Pr
3610  FNEND
3620  DEF FNK(Cf,T)
3630  IF Cf=1 THEN
3640! RANGE: 30 TO 90 DEG C
3650  K=8.2095238E-2-T*(2.2214286E-4+T*2.3809524E-8)
3660  END IF
3670  IF Cf=2 THEN
3680  X=(T+273.15)/273.15
3690  K=-.92247+X*(2.8395-X*(1.8007-X*(.52577-.07344*X)))
3700  END IF
3710  RETURN K
3720  FNEND
3730  DEF FNTanh(X)
3740  P=EXP(X)
3750  Q=EXP(-X)
3760  Tanh=(P-Q)/(P+Q)
3770  RETURN Tanh
3780  FNEND
3790  DEF FNTvsv(V)
3800  COM /Cc/ C(7)
3810  T=C(0)
3820  FOR I=1 TO 7
3830  T=T+C(I)*V^I
3840  NEXT I
3850  T=T+8.34643E-1-T*(2.4547896E-2-T*1.3722583E-4)
3860  RETURN T
3870  FNEND
3880  DEF FNBeta(Cf,T)
3890  Rop=FNRho(Cf,T+.1)
3900  Rom=FNRho(Cf,T-.1)
3910  Beta=-2/(Rop+Rom)*(Rop-Rom)/.2
3920  RETURN Beta
3930  FNEND
3940  DEF FNTsat(Cf,P)
3950  Tl=10
3960  Th=110
```

```
3970  Ta=(Tl+Th)*.5
3980  Pc=FNPsat(Cf,Ta)
3990  IF ABS((P-Pc)/P)>.0001 THEN
4000  IF Pc>P THEN Th=Ta
4010  IF Pc<P THEN Tl=Ta
4020  GOTO 3970
4030  END IF
4040  RETURN Ta
4050  FNEND
4060  DEF FNPsat(Cf,Tc)
4061  IF Cf=1 THEN
4070  T=Tc*1.8+32+459.6
4080  P=10^(33.0655-4330.98/T-9.2635*LGT(T)+2.0539E-3*T)
4081  END IF
4082  IF Cf=2 THEN
4083  P=Pvstw(Tc)
4085  END IF
4090  RETURN P
4100  FNEND
4110  SUB Plot
4120  DIM C(9)
4130  INTEGER Ii
4140  PRINTER IS 1
4150  Idv=1
4160  BEEP
4170  INPUT "LIKE DEFAULT VALUES FOR PLOT (1=Y,0=N)?",Idv
4180  BEEP
4190  PRINT USING "4X,""Select Option:"""
4200  PRINT USING "4X,""0  q versus delta-T"""
4210  PRINT USING "4X,""1  h versus delta-T"""
4220  PRINT USING "4X,""2  h versus q"""
4230  PRINT USING "4X,""3  h-ratio versus delta-T"""
4240  INPUT Opo
4250  IF Opo=3 THEN
4260  BEEP
4270  INPUT "SELECT TUBE DIAMETER (0=.75,1=1.0 IN)",Itd
4280  END IF
4290  PRINTER IS 705
4300  IF Idv<>1 THEN
4310  BEEP
4320  INPUT "ENTER NUMBER OF CYCLES FOR X-AXIS",Cx
4330  BEEP
4340  INPUT "ENTER NUMBER OF CYCLES FOR Y-AXIS",Cy
4350  BEEP
4360  INPUT "ENTER MIN X-VALUE (MULTIPLE OF 10)",Xmin
4370  BEEP
4380  INPUT "ENTER MIN Y-VALUE (MULTIPLE OF 10)",Ymin
4390  ELSE
4400  Cy=3
4410  IF Opo=0 THEN
4420  Cx=2
4430  Xmin=1
4440  Ymin=1000
4450  END IF
4460  IF Opo=1 THEN
4470  Cx=2
4480  Xmin=1
4490  Ymin=100
4500  END IF
4510  IF Opo=2 THEN
4520  Cx=3
```

```
4530  Xmin=1000
4540  Ymin=100
4550  END IF
4560  IF Opo=3 THEN
4570  Cx=2
4580  Cy=3
4590  Xmin=1
4600  Ymin=1
4610  END IF
4620  END IF
4630  BEEP
4640  PRINT "IN;SP1;IP 2300,1800,8300,6800;"
4650  PRINT "SC 0,100,0,100;TL 2,0;"
4660  Sfx=100/Cx
4670  Sfy=100/Cy
4680  PRINT "PU 0,0 PD"
4690  Nn=9
4700  FOR I=1 TO Cx+1
4710  Xat=Xmin*10^(I-1)
4720  IF I=Cx+1 THEN Nn=1
4730  FOR J=1 TO Nn
4740  IF J=1 THEN PRINT "TL 2 0"
4750  IF J=2 THEN PRINT "TL 1 0"
4760  Xa=Xat*J
4770  X=LGT(Xa/Xmin)*Sfx
4780  PRINT "PA";X;",0; XT;"
4790  NEXT J
4800  NEXT I
4810  PRINT "PA 100,0;PU;"
4820  PRINT "PU PA 0,0 PD"
4830  Nn=9
4840  FOR I=1 TO Cy+1
4850  Yat=Ymin*10^(I-1)
4860  IF I=Cy+1 THEN Nn=1
4870  FOR J=1 TO Nn
4880  IF J=1 THEN PRINT "TL 2 0"
4890  IF J=2 THEN PRINT "TL 1 0"
4900  Ya=Yat*J
4910  Y=LGT(Ya/Ymin)*Sfy
4920  PRINT "PA 0,";Y;"YT"
4930  NEXT J
4940  NEXT I
4950  PRINT "PA 0,100 TL 0 2"
4960  Nn=9
4970  FOR I=1 TO Cx+1
4980  Xat=Xmin*10^(I-1)
4990  IF I=Cx+1 THEN Nn=1
5000  FOR J=1 TO Nn
5010  IF J=1 THEN PRINT "TL 0 2"
5020  IF J>1 THEN PRINT "TL 0 1"
5030  Xa=Xat*J
5040  X=LGT(Xa/Xmin)*Sfx
5050  PRINT "PA";X;",100; XT"
5060  NEXT J
5070  NEXT I
5080  PRINT "PA 100,100 PU PA 100,0 PD"
5090  Nn=9
5100  FOR I=1 TO Cy+1
5110  Yat=Ymin*10^(I-1)
5120  IF I=Cy+1 THEN Nn=1
5130  FOR J=1 TO Nn
```

```
5140  IF J=1 THEN PRINT "TL 0 2"
5150  IF J>1 THEN PRINT "TL 0 1"
5160  Ya=Yat*J
5170  Y=LGT(Ya/Ymin)*Sfy
5180  PRINT "PD PA 100.",Y,"YT"
5190  NEXT J
5200  NEXT I
5210  PRINT "PA 100,100 PU"
5220  PRINT "PA 0,-2 SR 1.5,2"
5230  I1=LGT(Xmin)
5240  FOR I=1 TO Cx+1
5250  Xa=Xmin*10^(I-1)
5260  X=LGT(Xa/Xmin)*Sfx
5270  PRINT "PA";X,",0;"
5280  IF I1>=0 THEN PRINT "CP -2,-2;LB10;PR -2,2;LB";I1;""
5290  IF I1<0 THEN PRINT "CP -2,-2;LB10;PR 0,2;LB";I1;""
5300  I1=I1+1
5310  NEXT I
5320  PRINT "PU PA 0,0"
5330  I1=LGT(Ymin)
5340  Y10=10
5350  FOR I=1 TO Cy+1
5360  Ya=Ymin*10^(I-1)
5370  Y=LGT(Ya/Ymin)*Sfy
5380  PRINT "PA 0,";Y,""
5390  PRINT "CP -4,-.25;LB10;PR -2,2;LB";I1;""
5400  I1=I1+1
5410  NEXT I
5420  IF Idv<>1 THEN
5430  BEEP
5440  INPUT "ENTER X-LABEL",Xlabel$
5450  BEEP
5460  INPUT "ENTER Y-LABEL",Ylabel$
5470  ELSE
5480  IF Opo=0 THEN
5490  Xlabel$="Tw-Tsat (K)"
5500  Ylabel$="q (W/m 2)"
5510  END IF
5520  IF Opo=1 THEN
5530  Xlabel$="Tw-Tsat (K)"
5540  Ylabel$="h (W/m 2.K)"
5550  END IF
5560  IF Opo=2 THEN
5570  Xlabel$="q (W/m 2)"
5580  Ylabel$="h (W/m 2.K)"
5590  END IF
5600  IF Opo=3 THEN
5610  Xlabel$="Tw-Tsat (K)"
5620  Ylabel$="h(enh)/h(smooth)"
5630  END IF
5640  END IF
5650  PRINT "SR 1.5,2;PU PA 50,-16 CP";-LEN(Xlabel$)/2;"0;LB";Xlabel$;""
5660  PRINT "PA -14,50 CP 0,";-LEN(Ylabel$)/2*5/6;"DI 0,1;LB";Ylabel$;""
5670  PRINT "CP 0,0 DI"
5680 Repeat:!
5690  BEEP
5700  INPUT "WANT TO PLOT DATA FROM A FILE (1=Y,0=N)?",Ok
5710  IF Ok=1 THEN
5720  BEEP
5730  INPUT "ENTER THE NAME OF THE DATA FILE",D_file$
5740  ASSIGN @File TO D_file$
```

```
5750  BEEP
5760  BEEP
5770  INPUT "ENTER THE BEGINNING RUN NUMBER",Md
5780  BEEP
5790  INPUT "ENTER THE NUMBER OF X-Y PAIRS STORED",Npairs
5800  BEEP
5810  PRINTER IS 1
5820  PRINT USING "4X,""Select a symbol:"""
5830  PRINT USING "4X,""1  Star      2  Plus sign"""
5840  PRINT USING "4X,""3  Circle    4  Square"""
5850  PRINT USING "4X,""5  Rombus"""
5860  PRINT USING "4X,""6  Right-side-up triangle"""
5870  PRINT USING "4X,""7  Up-side-down triangle"""
5880  INPUT Sym
5890  PRINTER IS 705
5900  PRINT "PU DI"
5910  IF Sym=1 THEN PRINT "SM*"
5920  IF Sym=2 THEN PRINT "SM+"
5930  IF Sym=3 THEN PRINT "SMo"
5940  IF Md>1 THEN
5950  FOR I=1 TO (Md-1)
5960  ENTER @File;Ya,Xa
5970  NEXT I
5980  END IF
5990  FOR I=1 TO Npairs
6000  ENTER @File;Ya,Xa
6010  IF Opo=1 THEN Ya=Ya/Xa
6020  IF Opo=2 THEN
6030  Q=Ya
6040  Ya=Ya/Xa
6050  Xa=Q
6060  END IF
6070  IF Opo=3 THEN Ya=Ya/FNHsmooth(Xa,Itd)
6080  X=LGT(Xa/Xmin)*Sfx
6090  Y=LGT(Ya/Ymin)*Sfy
6100  IF Sym>3 THEN PRINT "SM"
6110  IF Sym<4 THEN PRINT "SR 1.4,2.4"
6120  PRINT "PA",X,Y,""
6130  IF Sym>3 THEN PRINT "SR 1.2,1.6"
6140  IF Sym=4 THEN PRINT "UC2,4,99,0,-8,-4,0,0,8,4,0;"
6150  IF Sym=5 THEN PRINT "UC3,0,99,-3,-6,-3,6,3,6,3,-6;"
6160  IF Sym=6 THEN PRINT "UC0,5,3,99,3,-8,-6,0,3,8;"
6170  IF Sym=7 THEN PRINT "UC0,-5,3,99,-3,8,6,0,-3,-8;"
6180  NEXT I
6190  BEEP
6200  ASSIGN @File TO *
6210  GOTO 5690
6220  END IF
6230  PRINT "PU SM"
6240  BEEP
6250  INPUT "WANT TO PLOT A POLYNOMIAL (1=Y,0=N)?",Go_on
6260  IF Go_on=1 THEN
6270  BEEP
6280  INPUT "ENTER LOWER AND UPPER X-LIMITS",Xll,Xlu
6290  FOR Xx=0 TO Cx STEP Cx/200
6300  Xa=Xmin*10^Xx
6310  IF Xa<Xll OR Xa>Xlu THEN 6380
6320  Ya=FNPoly(Xa)
6330  Y=LGT(Ya/Ymin)*Sfy
6340  X=LGT(Xa/Xmin)*Sfx
6350  IF Y<0 THEN Y=0
```

```
6360  IF Y>100 THEN GOTO 6380
6370  PRINT "PA",X,Y,"PD"
6380  NEXT Xx
6390  END IF
6400  PRINT "PU PA 0,0 SP0"
6410  SUBEND
6420  DEF FNHsmooth(X,Itd)
6430  Hs=FNPoly(X)/X
6440  IF Itd=1 THEN Hs=Hs*.83347
6450  RETURN Hs
6460  FNEND
6470  DEF FNPoly(X)
6480  Poly=-4.4123718E+2-X*(6.8123917E+2-X*3.7416863E+2)
6490  RETURN Poly
6500  FNEND
6510  DEF FNPvst(Tsteam)
6515  DIM K(8)
6520  DATA -7.691234564,-26.08023696,-168.1706546,64.23285504,-118.9646225
6525  DATA 4.16711732,20.9750676,1.E9,6
6530  READ K(*)
6535  T=(Tsteam+273.15)/647.3
6540  Sum=0
6545  FOR N=0 TO 4
6550  Sum=Sum+K(N)*(1-T)^(N+1)
6555  NEXT N
6560  Br=Sum/(T*(1+K(5)*(1-T)+K(6)*(1-T)^2))-(1-T)/(K(7)*(1-T)^2+K(8))
6565  Pr=EXP(Br)
6570  P=22120000*Pr
6575  RETURN P
6580  FNEND
```

APPENDIX B

SAMPLE CALCULATIONS

Data set number fifteen of file GT6H29 corresponding to tube number 6 was chosen for the sample calculation.

A. TEST-SECTION DIMENSIONS

O.D.	=	25.30 (mm)
H_F	=	1.05 (mm)
fins/mm	=	1.02
inter-fin gap	=	0.25 (mm)
D_2	=	23.10 (mm)
D_1	=	17.86 (mm)
D_o	=	22.22 (mm)
D_i	=	20.32 (mm)
L_U	=	31.75 (mm)
L	=	50.80 (mm)

B. MEASURED PARAMETERS

V	=	70.0 (volts)
V_{RES}	=	4.8 (volts)
R	=	2.031 (ohms)
k_C	=	383 (W/m.K)
T_1	=	54.23 (°C)
T_2	=	54.29 (°C)
T_3	=	53.91 (°C)

T_4 = 53.65 (°C)

T_5 = 53.40 (°C)

T_6 = 53.31 (°C)

T_7 = 53.22 (°C)

T_8 = 53.54 (°C)

T_{SAT} = 47.51 (°C)

C. PROPERTIES OF FREON-113 AT SATURATION TEMPERATURE

μ = 4.948×10^{-4} (kg/m.s)

ρ = 1510.0 (kg/m^3)

C_P = 0.973 (kJ/kg.K)

k = 0.0715 (W/m.K)

D. HEAT-FLUX CALCULATION

The heat transfer rate from the cartridge heater:

$$Q_H = VI \text{ (W)} \quad \text{and} \quad I = V_{RES}/R \text{ (amps)}$$

$$I = 4.8/2.031 = 2.363 \text{ (amps)}$$

$$Q_H = 70.0 \times 2.363 = 165.410 \text{ (WO}$$

The cross-sectional area of machined ends:

$$A_C = \pi/4.0(D_o^2 - D_i^2) \text{ (m}^2\text{)}$$

$$A_C = \pi/4.0(0.02222^2 - 0.02032^2) = 63.481 \times 10^{-6} \text{ (m}^2\text{)}$$

The tube outside wall perimeter:

$$P = \pi D_o = 3.1416 \times 0.02222 = 6.981 \times 10^{-2} \text{ (m)}$$

The corrected length of ends:

$$L_C = L_U + t/2.0 \qquad t = D_o - D_i$$

$$L_C = 0.03175 + (0.02222 - 0.02032)/2.0 = 32.7 \times 10^{-3} \text{ (m)}$$

Other thermophysical properties:

$$\nu = \mu/\rho = 4.948 \times 10^{-4}/1510.0 = 3.277 \times 10^{-7} \text{ (m}^2\text{/s)}$$

$$\alpha = k/\rho C_P = 0.0715/(1510.0 \times 973) = 4.866 \times 10^{-8} \text{ (m}^2\text{/s)}$$

$$\beta = -\frac{1}{\rho}\frac{\Delta\rho}{\Delta T} = -\frac{1}{1510}\left(\frac{1509.77 - 1510.26}{47.61 - 47.41}\right)$$

$$= 0.00162 \text{ (1/K)}$$

$$Pr = \nu/\alpha = 3.277 \times 10^{-7}/4.866 \times 10^{-8} = 6.734$$

Temperature at the base of the T-shaped fins:

$$T_B = T_{AVG} - Q_H \frac{\ln(D_2/D_1)}{2\pi L k_C}$$

$$T_{AVG} = \left(\sum_{n=1}^{8} T_N\right)/8.0 = 53.69 \text{ (°C)}$$

$$T_B = 53.69 - 165.41 \frac{\ln(23.1/17.86)}{2\pi \times 0.0508 \times 383} = 53.34 \text{ (°C)}$$

$$\theta_B = T_B - T_{SAT} = 53.342 - 47.51 = 5.832 \text{ (K)}$$

Average heat-transfer coefficient at the unenhanced ends:

$$\bar{h} = \frac{k}{D_o}\left\{0.60 + 0.387\,\frac{\left[\dfrac{g\beta D_o^3\theta_B \operatorname{Tanh}\left\{\left(\dfrac{\bar{h}P}{k_C A_C}\right)^{1/2} L_C\right\}}{\nu\alpha L_C\left(\dfrac{\bar{h}P}{k_C A_C}\right)^{1/2}}\right]^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}}\right\}^2$$

$$\bar{h} = 187.49 \text{ (W/m}^2\text{.K)}$$

Heat-transfer rate through unenhanced ends:

$$Q_F = \sqrt{\bar{h}\,P\,k_C\,A_C}\;\theta_B \operatorname{Tanh} mL_C$$

$$m = \left(\frac{\bar{h}P}{k_C A_C}\right)^{1/2} = \left(\frac{187.489 \times 6.981 \times 10^{-2}}{0.0715 \times 63.481 \times 10^{-6}}\right)^{1/2} = 1.698 \times 10^3$$

$$Q_F = 3.29 \text{ (W)}$$

$$Q_{LOSS} = 2Q_F = 6.579 \text{ (W)}$$

Heat flux through Gewa-T surface:

$$Q = Q_H - Q_{LOSS}$$

$$Q = 165.41 - 6.579 = 158.83 \text{ (W)}$$

$$q = Q/A_B$$

$$A_B = \pi \times 23.1 \times 10^{-3} \times 50.8 \times 10^{-3} = 3.687 \times 10^{-3} \text{ (m}^2\text{)}$$

$$q = \frac{158.83}{3.687 \times 10^{-3}} = 43.978 \times 10^3 \text{ (W/m}^2\text{)}$$

The following are the results obtained from the computer by running the data reduction program (DRP) (see Appendix A):

$$Q = 44.466 \quad 10 \text{ (W/m}^2\text{)}$$

$$\theta_B = 5.76 \text{ (K)}$$

$$h = 187.39 \text{ (W/m}^2\text{.K)}$$

The small difference in results obey the inaccuracy of hand calculations with the use of about three significant digits only.

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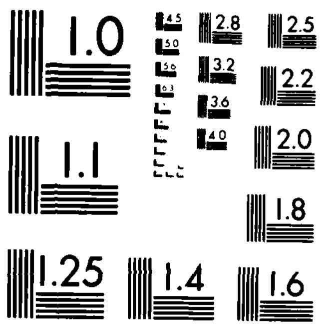

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APPENDIX C

UNCERTAINTY ANALYSIS

The same data set (set No. 15 of file GT6H29) that was used for the sample calculation was chosen for the uncertainty analysis; therefore dimensions of test section, measured and calculated parameters found in sample calculation were used in this analysis. All uncertainties are presented as a percentage of the calculated parameter.

A. UNCERTAINTY IN SOURCE HEAT-TRANSFER RATE

$$Q = VI \ (W), \qquad I = V_{RES}/R \ (amp)$$

$$\delta V = \delta V_{RES} = 0.05 \ (volts)$$

where:

δ = uncertainty in measurement and calculation

$$\delta I = \delta V_{RES}/R = 0.05/2.031 = 0.025 \ (amp)$$

$$\frac{\delta Q_H}{Q_H} = \left[\left(\frac{\delta V}{V}\right)^2 \quad \left(\frac{\delta I}{I}\right)^2\right]^{1/2}$$

$$\frac{\delta Q_H}{Q_H} = \left[\left(\frac{0.05}{70}\right)^2 + \left(\frac{0.025}{2.363}\right)^2\right]^{1/2} = 1.06 \text{ percent}$$

B. UNCERTAINTY IN SURFACE AREA

$$A_B = \pi D_2 L \qquad \delta D_2 = \delta_L = 0.1 \text{ (mm)}$$

$$\frac{\delta A_B}{A_B} = [(\frac{0.1}{23.1})^2 + (\frac{0.1}{50.8})^2]^{1/2} = 0.48 \text{ percent}$$

C. UNCERTAINTY IN ΔT

$$\Delta T = T_W - T_{SAT} \qquad \delta T_W = \delta T_{SAT} = 0.15 \text{ (°C)}$$

$$\frac{\delta \Delta T}{\Delta T} = [(\frac{\delta T_W}{\Delta T})^2 + (-\frac{\delta T_{SAT}}{\Delta T})^2]^{1/2}$$

$$\frac{\delta \Delta T}{\Delta T} = [(\frac{0.15}{6.18})^2 + (-\frac{0.15}{6.18})^2]^{1/2} = 3.43 \text{ percent}$$

D. UNCERTAINTY IN HEAT FLUX

$$q = \frac{Q_H - 2Q_F}{A_B}$$

$$\frac{\delta q}{q} = [(\frac{\delta Q_H}{Q_H - 2Q_F})^2 + (\frac{2\delta Q_F}{Q_H - 2Q_F})^2 + (-\frac{\delta A_B}{A_B})^2]^{1/2}$$

In this case, $Q_H = 165.41$ (W) and $Q_F = 3.29$ (W)

Therefore, Q_F = 2.0 percent of Q_H. Assuming the same proportion in the uncertainty for Q_F:

$$\delta Q_F = 0.02\ \delta Q_H = 0.02 \times 1.753 = 0.035\ \text{(W)}$$

$$\frac{\delta q}{q} = [(\frac{1.753}{158.83})^2 + (-\frac{2 \times 0.035}{158.83})^2 + (-0.0048)^2]^{1/2}$$

$$\frac{\delta q}{q} = 1.26 \text{ percent}$$

E. UNCERTAINTY IN BOILING HEAT-TRANSFER COEFFICIENT

$$\bar{h} = \frac{q}{\Delta T}$$

$$\frac{\delta \bar{h}}{\bar{h}} = [(\frac{\delta q}{q})^2 + (-\frac{\delta \Delta T}{\Delta T})]^{1/2}$$

$$\frac{\delta \bar{h}}{\bar{h}} = [(0.0126)^2 + (-0.0343)^2]^{1/2} = 3.65 \text{ percent}$$

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